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REVIEW OF ENGINEERING PROGRESS ABROAD

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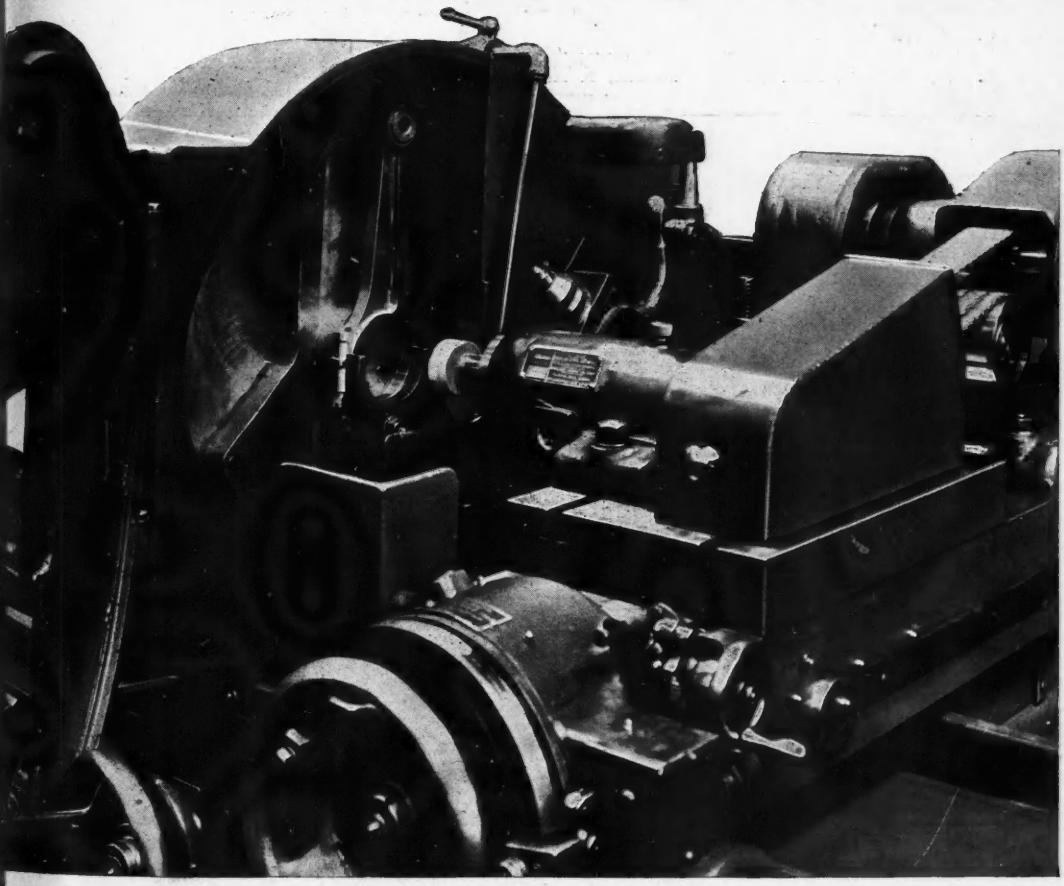

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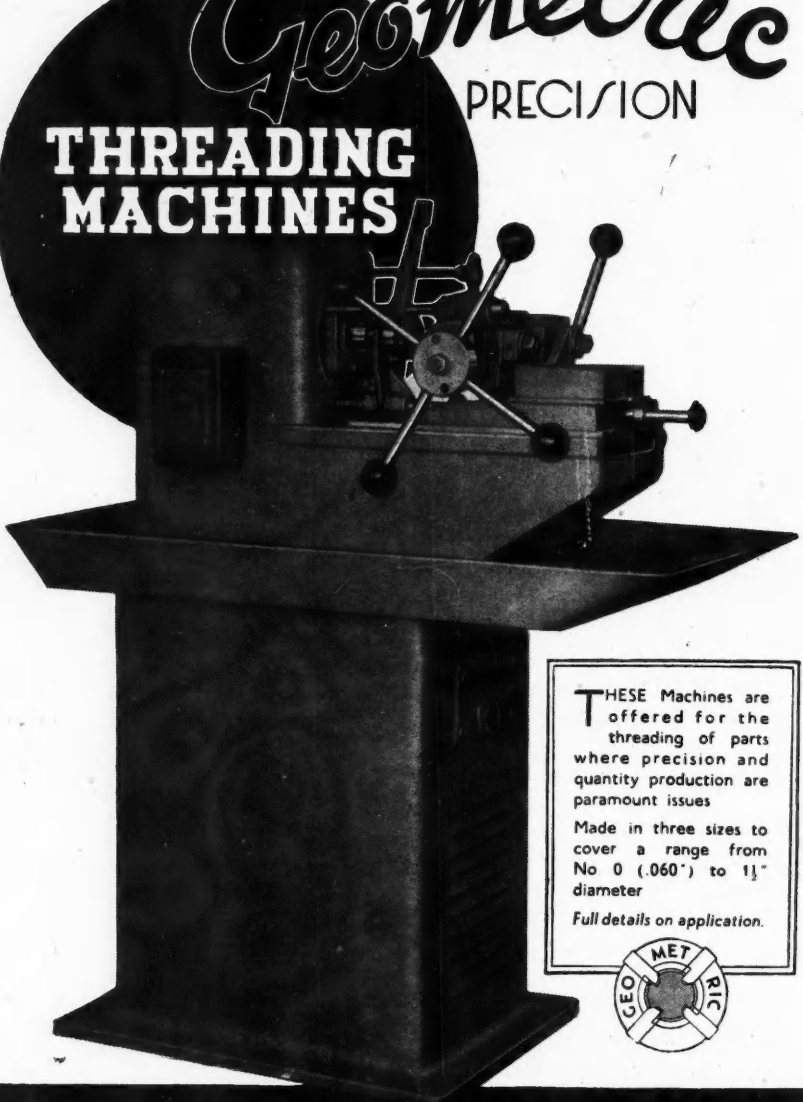

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PROBLEMS RELATED TO TWO-STROKE ENGINES

By Dr. Ing. RUDOLF MERTZ. (From *ATZ, Automobiltechnische Zeitschrift*, Vol. 45, No. 6, March, 1942, pp. 149–156).

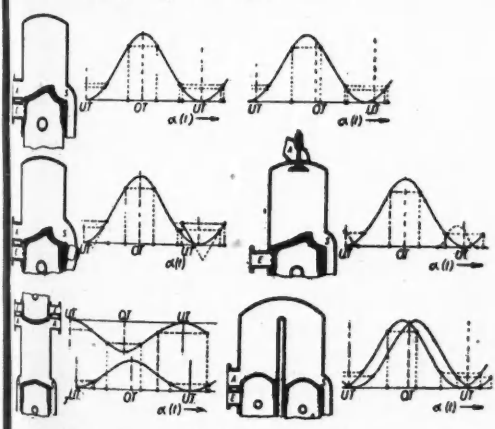
ALTHOUGH the development of two-stroke engines is almost as old as the four-stroke type, its adoption is not so universal as would be expected considering its simplicity of design and construction; some disadvantages impose a formidable handicap on two-stroke engines. Thus the efficiency of the usual three-channel type two-stroke engines is unsatisfactory because of the comparatively imperfect scavenging efficiency. The maximum pressure, too, is lower than in four-stroke types, for which the insufficient fresh air charge is responsible. It is, therefore, not surprising that the power developed in two-stroke engines is nothing like the double of the four-stroke type—under otherwise comparable conditions.

The technical difficulties are apparent from Fig.1, showing diagrammatically the cylinder of a two-stroke, three-port engine. Here the upper edge of the piston controls the opening and closing of the inlet and scavenging ports, while the exhaust port is controlled in a similar manner by the lower piston edge.

To prevent the products of combustion from entering into the crank case due to back pressure, the exhaust port must be opened before the scavenging port. But this means also that after closing the scavenging port, the closing of the exhaust port is delayed by the same amount and hence hinders supercharging of the cylinder. Furthermore, it is possible that part of the fresh air charge escapes through the exhaust channel.

The control of the inlet port, too, is unsatisfactory. For a considerable part of the upward stroke, the crank case is not in free communication with the ambient atmosphere. The pressure falls below atmospheric and eventually the fresh air will rush in at a high speed. For a similar period of the expansion stroke the air is permitted to escape from the crank case, thereby reducing the fresh air charge and the pressure in the crank case.

In the course of engine development a number of methods were tried to obtain an asymmetrical, i.e. improved, control of the port opening and closing. But almost every solution was based on the use of complicated machine parts and for this reason it is necessary to seek for yet another, simpler solution.



UT = O.D.C. ; OT = I.D.C.

Fig. 1. Valve diagram. Left : Standard arrangement ; Right : Crank axis inclined to cylinder axis.

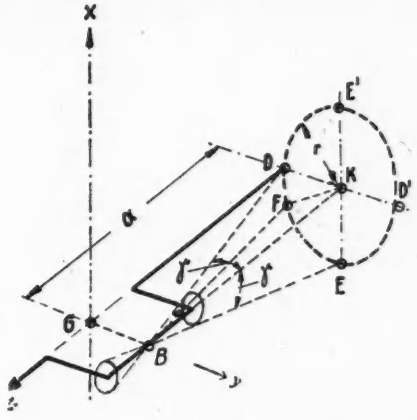


Fig. 2

As was mentioned before, the piston of the two-stroke engine serves not only as a means of transmitting the power, but also controls the opening and closing of the ports. A comparison with the sleeve valve type four-stroke engine is therefore pertinent. The original engine designed by Knight was fitted with two axially moving sleeve valves, one sleeve however, was found to do the job if in addition to the axial movement an oscillating rotating motion was provided.

Corresponding to this it appears that in the case of the two-stroke engine some improvements results should be obtained if the piston is given an oscillating rotating motion.

A very simple solution to this problem is given by skewing the crank-pin from the axial direction of the shaft. The connecting rod, of course, must be attached to the piston and the crank by a suitable cross-head arrangement. Finally the edge of the piston must be made sloping.

As to the skewing of the crank, there are several possibilities as shown in Fig. 2. The full line shows the crank and shaft in its original form. A circle of radius (r) is drawn at a distance (a) from the centre of the crank, so that the centre of the circle (k) is in the continuation of the crank axis. If now the axis of the crank is made to pass through D or D' we obtain the "inclined" crank arrangement. If the axis of the crank is passing through E or E', we get the "crossed" arrangement. Finally if it passes through any other point, say F we get yet another arrangement.

In all three cases, the piston performs an oscillating rotating motion in addition to the reciprocating motion. Assuming an indefinitely long connecting rod ($\lambda = 1 : \infty = 0$) the resultant motion follows a screw thread path. That is every point on the piston circumference describes a screwlike line during the upward (inward) stroke and returns along the same line. The envelope of this path is a straight line inclined at some angle to the horizontal (Fig. 3a). The effect of the finite length of the connecting rod is shown by the full line in Fig. 3a.

With the inclined crank type the line described is eliptical, being symmetrical if $\lambda = 0$. If $\lambda > 0$ an eliptical path is obtained which is, however, symmetrical only about the vertical axis (Fig. 3c). This means that during

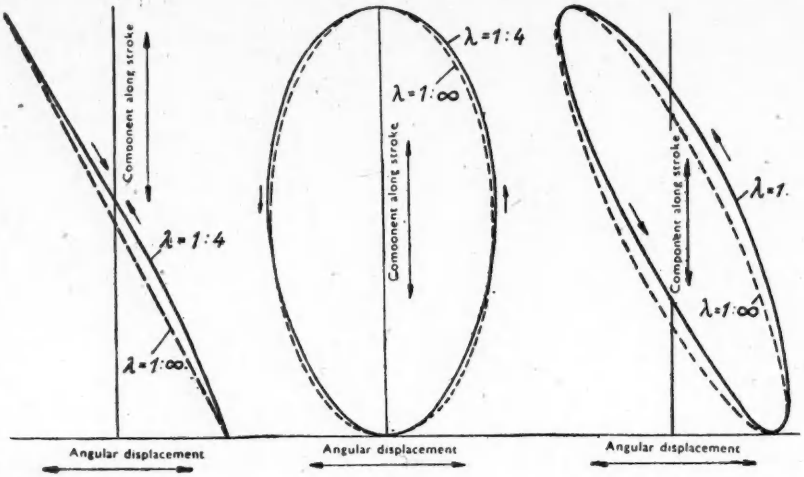


Fig. 3 *a* *b* *c*

the upward stroke a different path is described than during the downward stroke.

Finally with the continuation of the crank axis passing through (F) in Fig. 2, the path described by the piston circumference is intermediary between the above two types.

It will be clear from the foregoing that for our purpose the inclined crank type can be considered, as it is essential for proper port opening and closing that different paths be only described during the upward and downward stroke. Fig. 4 shows diagrammatically the arrangement of an inclined crank, two-stroke engine.

The value of (δ) which is the angular displacement of the piston in relation to the crank angle (α) can be estimated from Fig. 5. A plane passing through the crank DB and the connecting rod AB cuts the horizontal plane (y, z) along a line HD. The angle (δ) is the angle enclosed by the line HD and the crank shaft axis GD. The same angle is also included at $\widehat{EAF}$.

If (δ) is plotted as ordinate against the crank angle (α) as abscissæ the graph, Fig. 6, is obtained. This was actually plotted for the case when (δ) $=30°$ (Fig. 2).

Fig. 6 also shows that variation in the value of (λ) has very little effect on the value of (δ).

If now (δ) is plotted against the piston displacement, (h) as abscissa we obtain the curve shown in Fig. 7, which reminds us at once of the well known velocity diagram of the piston, along the axial direction.

The following mathematical formula enables the calculation of the value of (δ):—

$$\tan\delta = \tan\gamma\left(1 - \frac{\cos\alpha}{1/\lambda^2 - \sin^2\alpha}\right) \quad \ldots\ldots (1)$$

where α and γ are as before and $\lambda = r/l$.

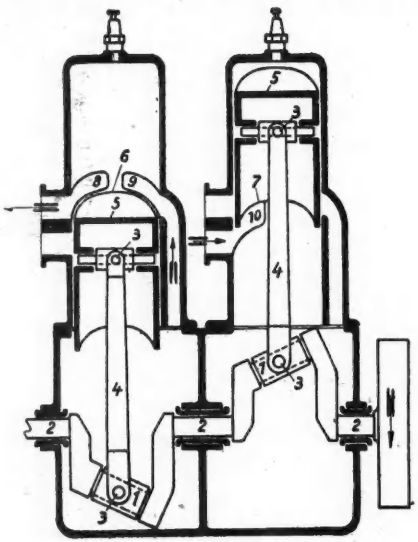


Fig. 4

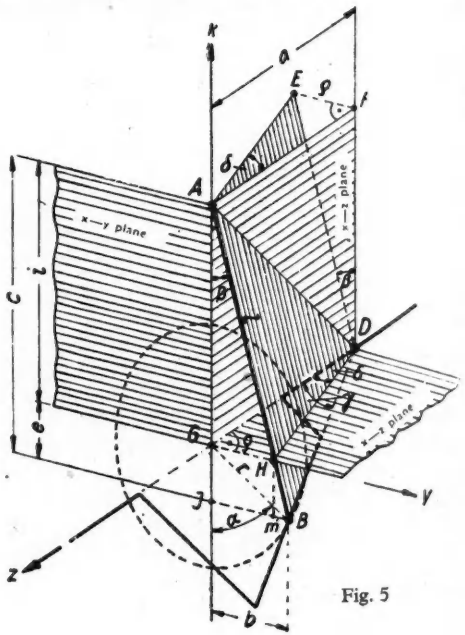


Fig. 5

A simpler equation for $\tan\delta$ is given by

$$\tan\delta = \tan\gamma\left(\sin\alpha - \frac{\lambda}{2}\cdot\sin 2\alpha\right) \quad \ldots\ldots (2)$$

which gives sufficiently accurate results for all practical purposes.

If $\alpha = 90°$, $\sin\alpha = 1$ and $\sin 2\alpha = 0$. Hence $\tan\delta_{(\alpha=90°)} = \tan\gamma$. This means in other words that when $\alpha = 90°$, δ becomes equal to γ which can be also seen from Fig. 5, 6 and 7.

If $\gamma = 1/\infty = 0$, $\tan\delta_{(\lambda=0)} = \tan\gamma\sin\alpha$ $\ldots\ldots (3)$

While the maximum value of δ for $\gamma = 0$ is $\delta_{max} = \gamma$, with a connecting rod of finite length this becomes

$$\tan\delta_{max} = \tan\gamma\cdot\sqrt{\lambda^2+1} \quad \ldots\ldots\ldots (4)$$

By writing $C = 2r/d$, the angular velocity of the rotating piston can be expressed as

$$\omega_{piston} = \frac{\tan\gamma}{C}\cdot\frac{\pi\cdot n}{30}\cdot\frac{\cos\alpha - \lambda\cos 2\alpha}{1+\left[\tan\gamma\left(\sin\alpha - \frac{\lambda}{2}\sin 2\alpha\right)\right]^2} \quad . (5)$$

where n = r.p.m. The accuracy of this equation may be considered as sufficient for general purposes

For the angular acceleration of the piston we have :—

$$\epsilon_{piston} = \frac{1}{C}\left(\frac{\pi\cdot n}{30}\right)^2\cdot\frac{w'' + w(w\cdot w'' - 2w'^2)}{1+w^2} \quad . (6)$$

where $w = \tan\delta = \tan\gamma\left(\sin\alpha - \frac{\lambda}{2}\sin 2\alpha\right)$

$$w' = \tan\gamma\,(\cos\alpha - \lambda\cos 2\alpha)$$

$$w'' = \tan\gamma\sin\alpha\,(4\lambda\cos\alpha - 1)$$

This equation also enables the calculation of the additional stress in the driving mechanism, which is of a very small order.

The value of (δ), angular velocity ω_{piston}, and angular acceleration ϵ_{piston}, are plotted against the stroke (8a) and the crank angle (8b) respectively.

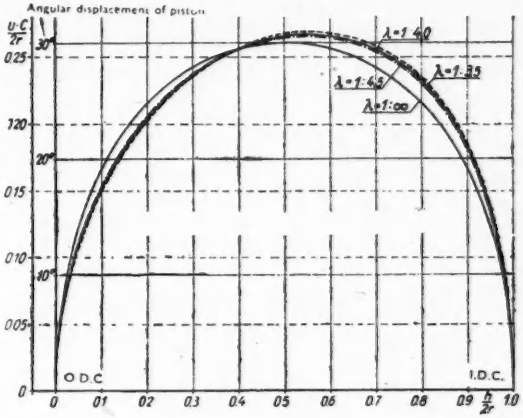


Fig. 7

It will now be described, how the oscillating rotating motion of the piston, together of course with the reciprocating motion can be used to obtain an asymmetrical control of the exhaust, inlet and scavenging port opening and closing. For simplicity's sake, let us assume that the inclination of the piston edge is uniform and is therefore as shown in Fig. 9a. The angle of pitch is $+\varphi$ and $-\varphi$ respectively at the two sides of the piston, where $+\varphi$ is not necessarily equal numerically to $-\varphi$. The shape of the piston is illustrated in Fig. 9b.

Owing to the rotating motion of the piston, a point on the circumference is displaced by an amount, u, (Fig. 9a) so that the position of the piston edge becomes now as shown by the dotted line. We have then :

$$u = \frac{\pi}{360}\,d\,.\,\delta \quad . . (7)$$

where d is the piston diameter and δ the angle of rotation of the piston. This equation can be written in a more convenient form :

$$u = \frac{\pi\,.\,r}{180}\cdot\frac{\delta}{C} \quad . . (8)$$

where r is the crank radius and $C = 2r/d$.

It can be seen from Fig. 9a that the rotation of the piston has the effect of an upward and downward motion of the piston such that the apparent stroke is given by :

$$f = u\,.\,\tan\varphi \quad . . (10)$$

Naturally the same argument holds also for the other side of

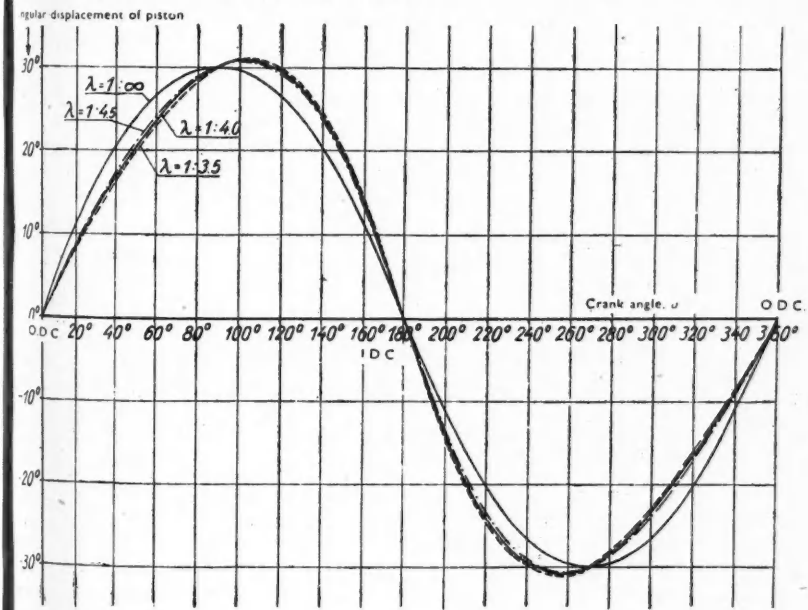


Fig. 6

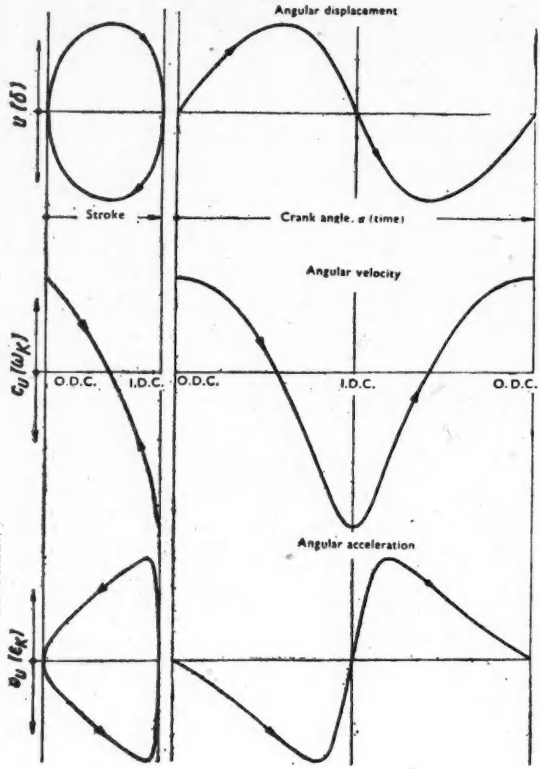


Fig. 8a Fig. 8b

the piston, where φ being negative, f becomes also negative. As the apparent stroke, f, is superimposed on the actual stroke of the piston, in effect it will cause a lead or a lag, according to whether f is positive or negative. The sum of the apparent stroke and actual stroke, h+f is h_e the effective stroke.

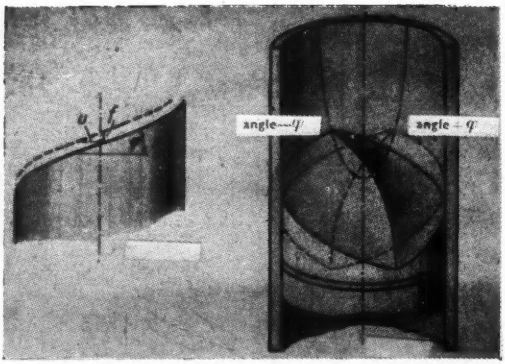


Fig. 9a Fig. 9b

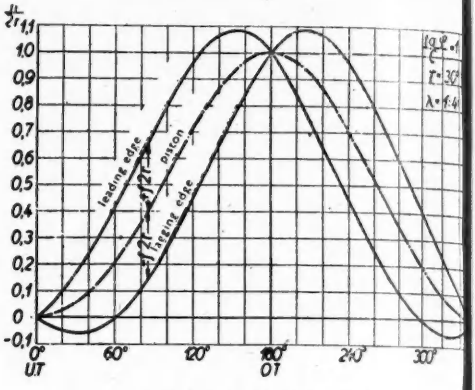


Fig. 10. (UT = O.D.C. ; OT = I.D.C.)

The displacement of both the leading and lagging edge is shown in Fig. 10. The diagram, Fig. 11, is very instructive too. It shows that the angle φ can be determined easily for any particular control of port opening or closing. In the example illustrated, the inlet port opens 76° before D.C., ($\alpha_1 = 104°$), inlet port closes 42° after D.C. ($\alpha_2 = 138°$). The value of f and h_e can also be determined from this diagram.

The right hand side of Fig. 11 shows the pumping effect in the crank casing. In a similar manner the p–v diagram could also be drawn. It appears that the pressure is reduced below atmospheric only for a very short period and compression starts much sooner than in standard engine types.

The practical use of the rotating piston can be illustrated by the following example. Assuming a value for $\gamma = 30°$, $\lambda = 1.4$, C = 1. the exhaust port is to open 65° before D.C., with an advance of 22°. Selecting two

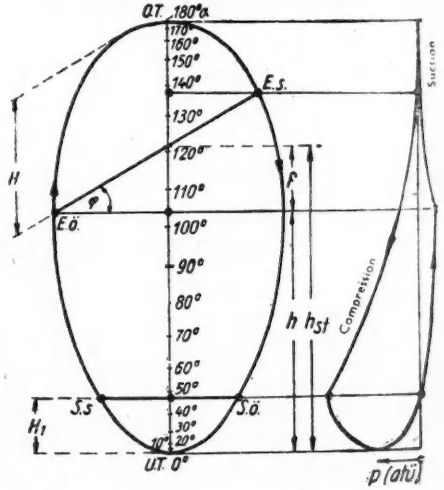


Fig. 11. (UT = O.D.C. ; OT = I.D.C. ; E.S. = Inlet port closed ; E.ö. = Inlet port open ; S.s = Suction port closed ; S.ö. = Suction port open.

values for φ, $\varphi = 20°$ and $\varphi = 30°$, we get four possibilities as shown in Table I. As any other set of two values could be selected, there are practically indefinite possibilities for controlling the port opening and closing.

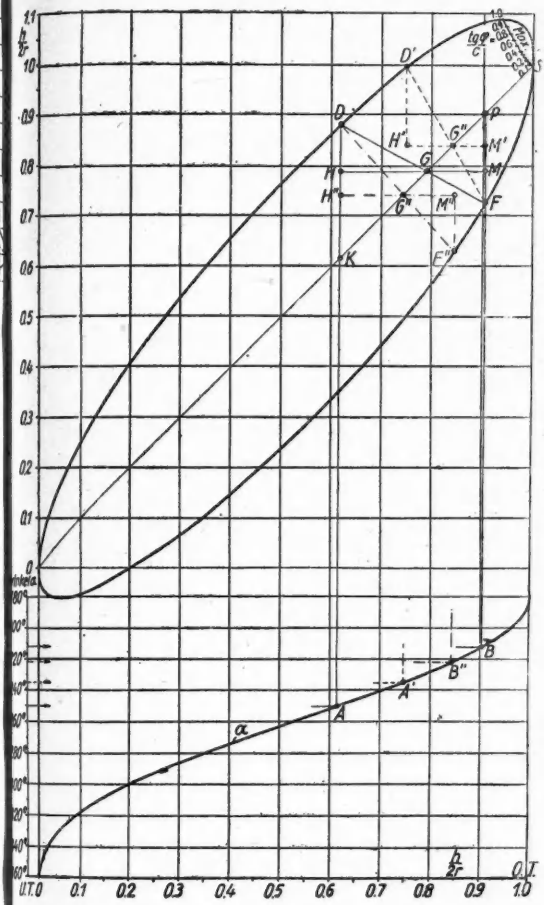


Fig. 12. (UT = O.D.C. ; OT = I.D.C.

TABLE I

	Example 1	Example 2	Example 3	Example 4
Inclination of Piston edge (φ)	$\varphi_1 = 20°$ $\varphi_2 = -20°$	$\varphi_1 = 30°$ $\varphi_2 = -30°$	$\varphi_1 = 30°$ $\varphi_2 = -20°$	$\varphi_1 = 20°$ $\varphi_2 = -30°$
Exhaust open	65° b.D.C.	65° b.D.C.	65° b.D.C.	65° b.D.C.
Exhaust closed	43° a.D.C.	29° a.D.C.	29° a.D.C.	43° a.D.C.
Scavenge open	43° b.D.C.	43° b.D.C.	43° b.D.C.	43° b.D.C.
Scavenge closed ..	65° a.D.C.	79° a.D.C.	65° a.D.C.	79° a.D.C.
Inlet port closing delayed	22°	50°	36°	36°
Height of ports :				
$H_{exh.}$ 1/2r ..	0.166	0.130	0.130	0.166
$H_{scav.}$ 1/2r..	0.166	0.220	0.166	0.220

Since quite apart from the crank angle, and the resulting angular displacement of the piston any desired value of the apparent stroke, f, can be obtained by selecting a suitable value for φ and C, then for the same value of $\tan\varphi/C$, f is also the same. This means that a certain effective stroke, h_e, can be achieved either by a larger piston diameter and flatter piston edge, or steeper piston edge and smaller diameter.

Assuming now $C = 2r/d = 1$, and $\varphi = 45°$ so that $\tan\varphi = 1$, then $\tan\varphi/C$ is also equal to 1. Calculating the values of (f) for these values, and plotting $h_e/2r$ against the abscissa h/2r we obtain Fig. 12. This diagram is particularly suited to find the value of $\tan\varphi/C$ for predetermined control of the ports. The drawing includes also the curve of the crank angle so that the effective stroke can be determined quickly for any crank angle position.

As an example, if it is required that the inlet port should open at 70° before D.C. and close 32° after D.C., then we get : $\alpha_1 = 110°$, and $\alpha_2 = 212°$, corresponding to points A and B on the α-curve. On the $\tan\varphi/C$ curve the corresponding points are D and F. A line joining D and F cuts the stroke line at G and a horizontal line through G meets the extension of AD and FB at H and M respectively. Since the apparent strokes DK and FP correspond to $\tan\varphi/C = 1$, the stroke HK and MP correspond to a value of $\tan\varphi/C$ which also satisfies the condition for α_1 and α_2 as above. Hence

$$\frac{\tan\varphi}{C} = \frac{HK}{DK} = \frac{MP}{FP} = 0.65 \quad \ldots\ldots \quad (10)$$

This value of $\tan\varphi/C$ can be obtained in more than one way. It must be remembered, however that once C has been fixed for one side of the piston, the same value of C must be used to determine $-\varphi$.

HOW SYNTHETIC RUBBERS MEET CURRENT DEMAND

By Otis D. Cole. (From *Machine Design*, Vol. 14, No. 7, July, 1942, pp. 76–81).

In the past few years the activity in rubber research laboratories all over the country has been increasing steadily and several materials have been developed which are approximately equal to natural rubber in most respects and for some purposes are better than the natural product. These materials are the result of systematic testing and selection of thousands of experimental materials in each class until only a few having the desired characteristics remain. Each of these was selected for commercial use because of certain outstanding characteristics which make it desirable. So far, the producers are to be commended for the relatively few materials which have been introduced to the industry. This fact is more readily appreciated when one considers that it is possible more or less to " tailor-make " a material for each special purpose.

There are numerous articles in the literature on the preparation and compounding of various types and varieties of synthetic rubbers and the characteristics of the products made of them. A comprehensive summary up to June, 1940, was given by Wood.[1]

It should be pointed out that the term " synthetic rubber " is actually a misnomer, because in order for the name to be correct the synthetic materials would have to

[1] L. A. Wood, *India Rubber World*, Vol. 102, No. 4, p. 33, 1940 ; *Rubber Chemistry Technology*, Vol. 13, p. 861, 1940

be identical chemically with natural rubber, which is not the case. These products should more properly be called "synthetic rubberlike materials." However, because of common usage they will be referred to in this paper as synthetic rubbers.

In addition to the class of synthetic rubberlike materials, there are available other materials which have some rubberlike characteristics. In this group are found organic polysulfide polymers (Thiokol), Vinylite, plasticised Polyvinyl Chloride (Koroseal), polybutylene (Vistanex), etc. These materials are thermoplastic even when compounded with other materials, instead of thermosetting as in the case of compounded rubber. The plasticised Vinylites have some properties such as chemical inertness which make them more desirable in cases where flexibility without resilience is required. The Thiokols or organic polysulfide polymers might be included among the synthetic rubbers but, although they appear to be vulcanisable, their thermoplasticity makes them useful only to a limited degree in service where elasticity and other rubberlike properties at elevated temperatures are desired. Thiokol is one of the oldest of the American types of synthetic materials having rubberlike properties. Its high resistance to oils, other solvents, and all kinds of ageing, as well as its impermeability to gases makes it serve a useful purpose in industry.

Four Types are Discussed.

Materials considered as synthetic rubbers and covered in the following fall into four classes: Neoprene types, Buna S types, Buna N types, and Butyl rubber.

Neoprene types are polymers of chloroprene or copolymers of chloroprene with other materials. It is evident that there is possible a wide variety of products all made of the same basic materials.

The term "neoprene", therefore, should not be considered as descriptive of a single product. Neoprenes as a class may be termed moderately oil-resisting rubbers.

Buna S types are copolymers of butadiene with styrene. A Buna S type of polymer has been adopted by the Rubber Reserve company for use in tyres and as a general purpose rubber. Fundamentally, it is less efficient and develops lower physical properties at ordinary temperatures than natural rubber, but it is much more heat resistant and when properly compounded is practically as cold resistant as natural rubber. It is similar to natural rubber in that it is not oil resistant.

Buna N types are copolymers of butadiene and acrylonitrile. They are generally oil and hydrocarbon solvent-resistant to a high degree; however, they are poor in resistance to oxygenated solvents. Their cold resistance in general is poor but, by correct compounding, fair to good cold resistance can be obtained. A large number of different varieties of this type are on the market under the trade-names of Hycar O.R., Chemigum I, II, III and X, Perbunan, Buna NM, NX, NXM, NXX, and NF, etc. These vary as to the ratios of the constituents and the method of preparation, but their properties as a class are similar. Butyl rubber is reported to be a copolymer of isobutylene with butadiene. It is high in chemical stability, very impermeable to gases and is said to be heat resistant. It may eventually have the lowest cost of all the synthetic rubbers. In the near future a new plant will go into production and at that time sufficient material will be made available, so that a complete evaluation of all its properties in all types of products can be made.

It should be understood that the possibility of a new material being found which surpasses all the present ones now in production or development is not remote, but rather is likely. The following examples of the more or less unusual properties of a few of the present materials are given with the foregoing in mind. New developments may greatly change the picture in the next few months.

When evaluating synthetic rubbers in the laboratory, it must be kept in mind that we are working with new, entirely different materials and not rubber. It is easy to make the mistake of attempting to interpret laboratory data on synthetic rubbers on the basis of experience with natural rubber, thus drawing erroneous conclusions.

Because of our present interest in the butadiene or Buna type synthetics, the majority of the examples will be chosen from that class. Each has been developed to fit a special need, such as ease of processing, ease of moulding, oil resistance, cold resistance, etc.

Data given in Table I show that the tensile strengths of pure gum stocks prepared from the Buna type rubbers are very low. It is only after these rubbers have been compounded with carbon black that they exhibit tensile properties which remotely resemble those of natural rubber stocks. Even then the tensile strengths are far below those of natural rubber stocks having similar compounding. Neoprene is different in this respect, for the pure gum stock has higher tensile strength than the stock compounded with channel black. It is evident that tensile strength is not the complete criterion for the judgment of quality of Buna-type synthetic rubbers, because if it were, none of them would have been considered further as possible materials for replacement of natural rubber in tyres, for example.

Tensile strengths of the various Buna N type synthetic rubbers show rather definite correlation with other physical properties such as oil resistance, cold resistance, etc., as will be shown later.

TABLE I

Tensile Data for Various Rubbers with Similar Compounding.

Tensile Strength (lbs./sq. in.)—Stress at 400% Elong. (lbs./sq. in.)—% Elong. at Break

Type of Rubber	Pure Gum	P–33* Black	Channel† Black	Pure Gum	P–33* Black	Channel† Black	Pure Gum	P–33* Black	Channel† Black
Natural ..	2500	—	4400	650	—	2100	800	—	600
Buna S. ..	200	1200	3000	—	1100	1900	300	430	580
Buna NM ..	700	1400	2500	550	1025	1450	420	520	560
Buna NXM ..	800	1600	2700	550	1225	1750	480	500	550
Buna NF ..	450	800	1300	—	825	—	300	400	340
Neoprene ..	3900	—	3400	950 (600%)	—	2200	1020	—	550

* These stocks were compounded with softeners, vulcanising ingredients, etc., and contained 100 parts of P–33 black per 100 parts of rubber or synthetic rubber.

† These stocks contained softeners, vulcanising ingredients and 50 parts of rubber grade channel black per 100 parts of rubber or synthetic rubber.

Resists Swelling in Oil.

The fact that the Buna N types of synthetic rubber exhibit very low swelling when exposed to oils is one of the most unusual properties and, at the same time, the most important property of this group. As a class they are better than neoprene in this respect. The different Buna N types have oil resistances as shown in Table II. Resistance to swelling in oil may be controlled, more or less, by proper selection of Buna N type synthetic rubbers. The negative swelling, or shrinking, of the Buna NXM stock in SAE 10 oil is most probably due to the extraction of some of the plasticiser by the oil. This property is important and must be compensated for by proper compounding when shrinkage is not desirable. By correct compounding it is possible to product a stock which will not change in volume when immersed in oil.

TABLE II.

Per Cent. Increase in Volume after 24 hours in different Oils.

Type of rubber	Paraffin Base SAE 10 Oil	80% Kerosene 20% Benzene
Buna S	152	—
Buna NM	3	16
Buna NXM	3	12
Buna NF	40	42
Neoprene GN ..	23	91

An interesting and rather unusual swelling phenomenon is illustrated by the following example: Two Buna NXM gasket compounds A and B were tested for resistance to swelling in carbon tetrachloride, which has a very pronounced swelling effect on all of the Buna N types. Compound A contained as the plasticiser an oil which was soluble, while compound B contained a resin which was practically insoluble in carbon tetrachloride. The data for per cent. swelling versus time of immersion for the two stocks are shown in Fig. 1. It is obvious that compound A would not be satisfactory as a gasket in contact with carbon tetrachloride. These two compounds in carbon tetrachloride are given as examples because the effect is exaggerated. However, our observations would indicate that this phenomenon is common, especially with aromatic solvents, and may be the cause of actual shrinkage in other oils. It is interesting to note that the usual three-day or seven-day immersion test would have rated these two compounds in resistance to swelling. The method of Garvey, lends itself more readily to following changes in swelling during immersion than does the regular A.S.T.M. method.

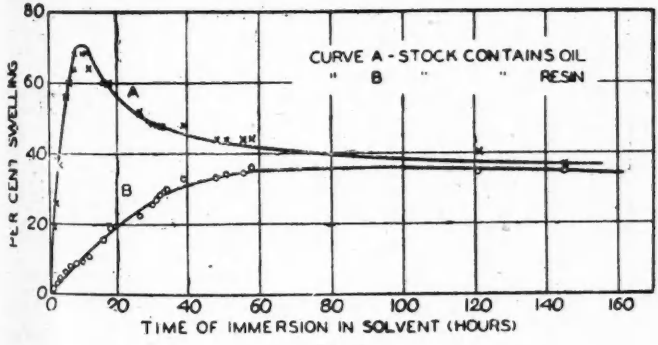


Fig. 1

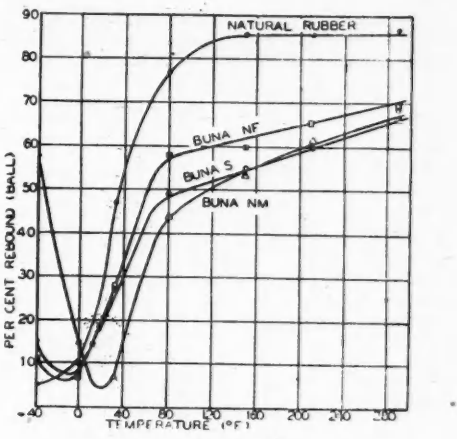


Fig. 2

The property of resilience is one which makes rubber an unusual material. In general, the Buna or Butadiene types of synthetic rubbers normally have lower resilience than does natural rubber, as shown in Fig. 2.

Compounding Increases Efficiency.

Although the data shown in Fig. 2 would indicate that the Buna S type of synthetic rubber is normally less efficient than natural rubber, it does not follow that it cannot be compounded to produce highly efficient stocks. However, when a stock is compounded to produce equal efficiency the hardness is increased over that of natural rubber compound. Fig. 3 shows a comparison between a high efficiency rubber compound and a Buna S compound designed to have approximately the same efficiency. The hardness values at the various temperatures are included to illustrate this point.

A comparison of the rebound efficiency of typical natural rubber, neoprene GN and Buna S high channel black compounds is shown in Fig. 4. These curves corroborate other efficiency tests which show that Buna S has higher hysteresis loss than natural rubber and neoprene GN.

Cold resistance, or the lack of it, in the oil-resistant synthetic rubbers is of utmost importance to those designing or specifying parts where these rubbers are to be used. The increase in resilience at low temperatures may be caused by changes in hardness as shown in Fig. 3. So far, no definite relationship between the shape of the resilience-temperature curve and the actual "brittle point" has been observed. It should be pointed out that the minimum point in the resilience-temperature curve does not mean that, at the temperature, the rubber is brittle and will break upon bending. There are some indications (see Fig. 3) that these minima mark the beginning of the definite stiffening or hardening which eventually renders the material unserviceable at lower temperatures.

In general, the cold resistance of the highly oil-resistant rubbers is poor—the more oil-resistant the

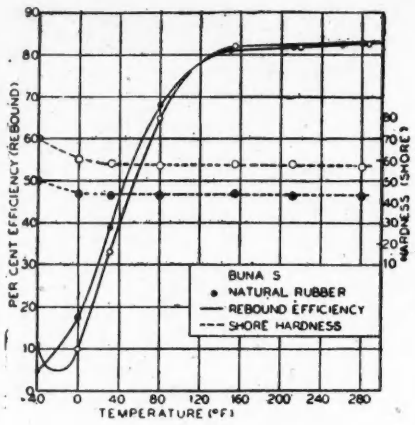


Fig. 3

rubber, the poorer the cold resistance. The data in Table III show the temperatures at which several Buna type and rubber compounds, varying only in the rubber used, become frozen or stiff to the touch. Refrigerated methyl alcohol was used as the coolant in this case. Values for the "brittle point" and "freezing point", taken from the literature, are given for comparison. All comparisons of data should be made within one column of the tabulation, not between different columns.

In certain cases where rubber parts are to be used in contact with a solvent in service, the cold resistance tests should be made using the same solvent, refrigerated, as the coolant. For example, a fuel pump diaphragm compound when tested in air was frozen solid at —15° F., but when immersed in the fuel it was perfectly flexible at —50° F. because of the plasticising action of the fuel.

The use of plasticisers plays an important part in the attainment of good cold resistance with the Buna N and neoprene types of synthetic rubbers. Fig. 5 shows the effect of various mixtures of two plasticisers A and B on the temperature at which several Buna NM and Buna NXM compounds ceased to become serviceable in air. It will be noted that Buna NM is more cold resistant in all cases than Buna NXM with the same plasticiser mixture present. It is possible to prepare compounds from Buna NF, treated with the proper plasticiser, which will be serviceable at temperatures as low as —70° F.

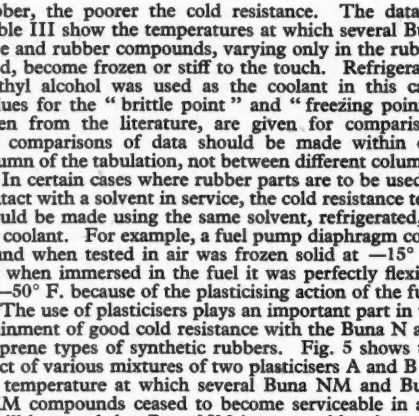


Fig. 4

A very outstanding and rather unusual property of most of the synthetic rubbers, especially those of the Buna and neoprene types, is their remarkable resistance to decomposition by heat. Thermal decomposition temperatures are shown in Table IV. The remarkably high heat resistance of Buna S as compared to natural rubber and the Buna N types would suggest its use for high temperature service where oil resistance is not a serious factor. Neoprene GN also shows superior heat resistance. However, all of the Buna and neoprene type synthetic rubbers become harder during continued exposure to high temperatures, but it is possible in some cases to plasticise them to overcome the hardening. This will tend to lower the heat resistance slightly.

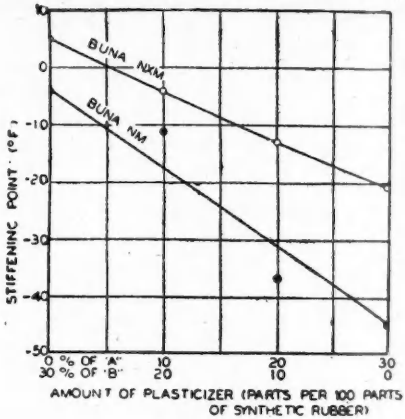


Fig. 5

TABLE III.

Stiffening, Freezing and Brittle Points.

Type of Rubber	Stiffening Point (°F.)	Brittle Point (°F.)	Freezing Point (°F.)
Natural ..	—48 to —52	—69.7	—86.8
Buna S ..	—36 to —40	—86.8 to —94	—86.8*
Buna NF..	—40	—	—
Buna NM	— 9	—51.7*	—
Buna NXM	—	—14.8to—18.4*	—
Neoprene GN ..	—33	—	—

* These stocks were undoubtedly prepared from different Buna type synthetic rubbers than those reported under "stiffening point."

TABLE IV.

Thermal Decomposition Temperatures.

Type of Rubber	Thermal Decomposition Temperature* (°C.)
Natural	194–198
Buna NF	220–229
Buna NM	220–229
Buna NXM	205–212
Buna S	Above 249
Neoprene GN	239–249

* Temperature at which a rubber compound will decompose in one hour.

Synthetic rubbers show exceedingly good resistance to all types of accelerated ageing. A comparison of the ageing characteristics of the various Buna types illustrated in Table I is given in Table V. Their resistance

to heat (air bomb) is remarkable when it is remembered that similar natural rubber compounds are badly deteriorated after only ten hours under the same conditions. As the data for oxygen bomb ageing indicate, the Buna N types show slightly less resistance to oxidation than the other types of ageing. However, their relative deterioration is much less than that of rubber under the same conditions. Buna S shows some susceptibility to oxidation, but is affected much less than natural rubber.

TABLE V.
Effect of Ageing.

Type of Rubber*	Per cent of Original Values. Air Bomb[1] Tensile	Elong.	Oxygen Bomb[2] Tensile	Elong.	Oven[3] Tensile	Elong.
Buna S ..	105	77	100	98	94	60
Buna NM ..	114	65	93	94	116	63
Buna NXM	112	64	103	94	118	68
Buna NF ..	151	70	106	83	150	73
Natural Rubber ..	—	—	56	—	91†	—

* Original values for these compounds are given under "P–33 Black" in Table I.
† After only 14 days in oven.
[1] Aged 15 hours in air bomb at 260° F, 80 lbs./sq. in. pressure.
[2] Aged 46 hours in oxygen bomb at 158° F, 300 lbs/sq. in. pressure.
[3] Aged 28 days in hot air oven at 158° F.

All the synthetic rubbers harden or become stiffer during heat treatments as illustrated by the elongation values in Table V. This fact must be considered when designs are being made for the use of these materials. All natural ageing data which have been obtained up to this time indicate that the Buna type synthetic rubbers will be entirely satisfactory in this respect.

Also no data for neoprene are given here, its high resistance to ageing is well known. However, it stiffens during heating and ageing to an equal if not greater degree than the Buna types.

Buna S types can be used in nearly all cases to replace natural rubber. Generally satisfactory service as compared to natural rubber has been observed in the case of passenger tyres. It must be pointed out that with this or any synthetic rubber, as in the case of any new material, certain difficulties are present and must be overcome. For example, truck and heavy-duty tyres at present are the result of years of specialised engineering and design, based on the characteristics of natural rubber. In order to meet the service conditions encountered to-day, radical departure from present thinking and designs may have to be made in order to fit synthetic rubber into the picture. It is sufficient to say that the Buna S types appear to fit more nearly into the scheme of things as we see them to-day than any of the other synthetic rubbers.

Buna N types of synthetic rubbers are being recommended for all types of mechanical goods such as hose, packings, gaskets and rollers, wherever high resistance to ageing, oils and hydrocarbon solvents is desired.

Neoprene types are used wherever moderate oil resistance and good resistance to ozone and sunlight is desired. Since it will not support combustion it is widely used as a flameproof, oilproof, wire insulation.

Butyl rubber can best be utilised where its inherent properties of chemical stability, resistance to ozone, light and high permeability to gases can be used to best advantage. As has been pointed out, no entirely satisfactory tyres have as yet been made from Butyl rubber.

Many of the really serious problems of to-day may be problems only because of the fact that certain handling processes designed for natural rubber are not fundamentally correct for the synthetic rubbers used. Because of this, it is possible that in the future the rubber industry will depend more than ever upon the mechanical engineer for the solution of its processing and handling problems.

STEEL-CLAD MERCURY RECTIFIERS WITH RARE GAS FILLING

By E. GERECKE, Geneva. (From *Bulletin of the Association Suisse des Electriciens*, Vol. 23, No. 8, April 22nd, 1942, pp. 226–230. Lecture read at a meeting of the S.E.V. on 21st June, 1941.)

FIG. 1 shows a pumpless rectifier designed by Secheron. All leads-in to the main anodes and the cathode, to the ignition and the excitation are placed into the lower part of the steel vessel (3). In consequence the mercury arc is short and the voltage drop low. The anodes (7) are protected from mercury drops by shields (6). The condensed mercury runs down into the drain (5) from where it flows back to the cathode through a tube. The upper part of the vessel the gas chamber (4) is filled with the rare gas when the rectifier is in operation. At low load the rare gas fills the whole vessel (3) in place of the mercury vapour then condensed.

Cooling is provided by air driven upwards around the vessel, by a fan arranged on the top. The cooling-air jacket (2) is earthed so that all live parts of the rectifier are protected. The terminals of the anodes (8) and of the cathode (9) are enclosed in the chamber (11) under the vacuum vessel. The whole apparatus is mounted on wheels.

The vessel is hermetically welded and tested by the covering outside with impregnated fabric and filling the inside with ammonia under pressure. The ammonia escaping by pores or cracks produces a change in the colour of the fabric. This test is so sensitive, that pores can be detected, which would not produce even within a year a measurable increase of the pressure inside the vessel. In order to drive off all gases from all parts of the

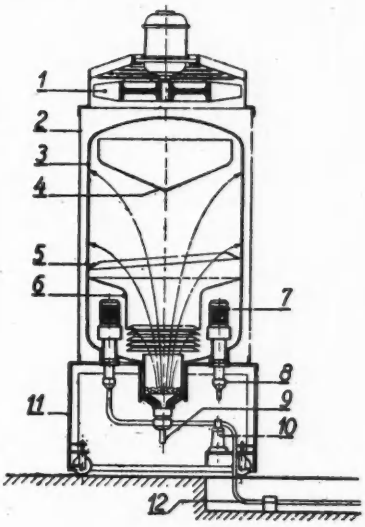


Fig. 1
Pumpless Mercury Rectifier Design Secheron (Sectional view).

rectifiers they are heated up to 400° to 500° C., and the gases set free from the walls are pumped out. Following this procedure, the anodes and the grids, which consist of graphite are degased by electric heating up to a temperature several hundred degrees higher than the normal working temperature. Any traces of gas which may be set free when the rectifier is working, are chemically combined with the iron of the vessel under the influence of the gas discharge. It has been found in Zurich, that hydrogen contained in the cooling water may enter the rectifier through the iron sheet of the rectifier vessel; therefore, water cooling has been replaced by air cooling. If through any reason hydrogen is liberated within the rectifier it is ionised by the arc and diffuses through the iron wall into the atmosphere.

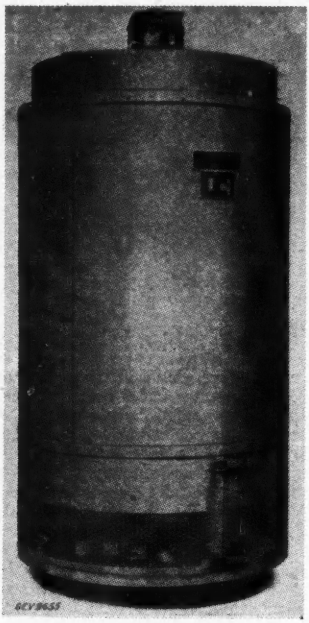


Fig. 2
Pumpless Rectifier Design Secheron.

The leads-in of the electrodes (Fig. 3), which have to stand the temperature of 400° to 500° C. during the formation, consist of a ceramic tube (1) on which two air tight metal collars are fixed. The lower collar (3) is welded to a branch (5) of the vacuum vessel (6), the other collar (2) to the conductor of the lead-in (4). For the fixing of the collars to the ceramic tube a thin layer of glass is used. The ceramic is heated in an electric oven with the two collars in right position. Glass, which is put between the tube and the collars, fuses and fills the narrow gap by capillary action. After cooling, the three materials ceramic, glass and metal, are in a perfectly tight connection. The coefficient of expansion of the metal has to be a little higher than that of the ceramic. In consequence the collars are shrunk on the tube after cooling. This has the advantage, that even in cases of quick heating, when the metal parts warm up faster than the insulators, no tensile stresses occur in the ceramic tube. The compressive strength of the ceramic is 10 to 20 times greater than the tensile strength. Moreover the tube is reinforced where the collars are fixed on it.

This method of connection of metal and ceramic has been tested by heating them up to 450° C., and then cooling to 20° C. and even to —40° C., such cycles being applied several hundred times. Ammonia tests showed that the connections which had undergone these temperature tests, were perfectly tight. The mechanical strength of the connection of the collars with the tube has been tested by drawing off the collars in axial direction by a press. The stress in the fused surface, when the collar yielded, has been found equal to the shearing strength of the glass.

An essential feature of the pumpless steel-clad rectifier is the filling with rare gas which became possible when no pump was used. This has turned out to be an unexpected advantage of the pumpless rectifier.

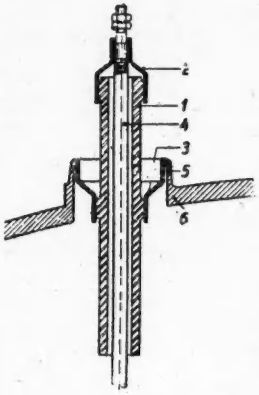


Fig. 3 Lead-in of an electrode.

The pressure of the mercury vapour in a rectifier depends on the temperature of the coolest part of the enclosure and equals to the corresponding pressure of saturation. The pressure in the rectifier when loaded must be higher than the pressure of saturation which corresponds to the temperature of the enclosing vessel, in order to start a transport of mercury from the space filled with mercury vapour to the wall of the vessel. The pressure may rise to the double or treble of the saturation pressure corresponding to the wall temperature. The saturation pressure decreases very rapidly with the temperature as can be seen from the following figures:

46°C.	p = 10.00	bar
17°C.	p = 1.00	„
— 7°C.	p = 0.10	„
—27°C.	p = 0.11	„

1 bar = 0.001 mm. Hg.

It has been found that the value of the current flowing through gas at low pressure cannot exceed a maximum value which depends on the cross-section and on the pressure; this behaviour of a gas at low pressure differs from that of metal conductor. This limit current rises with the temperature. A current greater than this limit current cannot flow, otherwise a higher voltage drop occurs and oscillations of medium frequency are produced: the rectifier and the transformer become a source of oscillations. The excess voltage may become as high as 40 kV., therefore, surge protective devices are connected to all anodes in rectifier plants. Oscillograms of the voltage drop between anodes and cathode may show oscillations of a frequency of 5 kHz and more. These excess voltages occur in cold rectifiers with currents far below rated load, and in warm vessels with overloads and short-circuits. Filling the vessels with rare gas of sufficient pressure eliminates these excess voltage phenomena. Therefore, such rectifiers may be loaded up to rated load or overload before they have attained normal working temperature. According to the draft of the I.E.C. Rules for rectifiers, 1939, overloads must be carried only at normal working temperature. The rectifier with rare gas filling can carry these overloads even when cold.

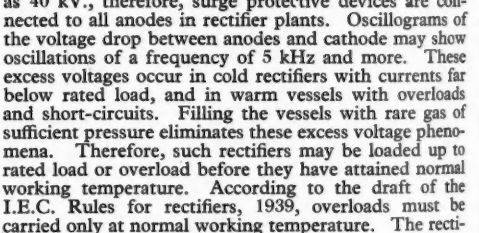
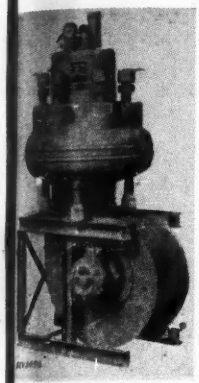
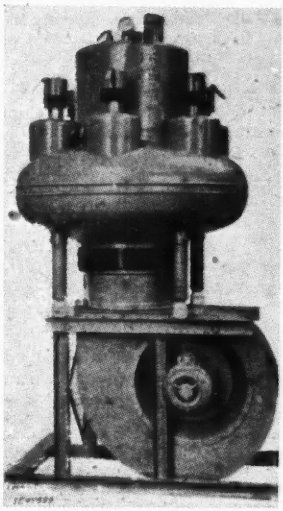


Fig. 4. Diagram showing the current delivered from a pumpless rectifier to a traction plant.





5. Pumpless Rectifier of
n Boveri, Type L 06 for
500 amp.
6. (Right): Pumpless
ifier Brown Boveri, Type
L 36 for 1200 amp.

DISCUSSION.

This report was followed by a discussion.

Ch. Ehrensperger, Brown Boveri, Baden, reported about pumpless rectifiers manufactured by Brown Boveri & Cie., Baden. This firm has been rather reserved in the beginning in regard to the problem of pumpless rectifiers. Meanwhile, however, the abolition of the pump has been recognised as an economical advantage, at least as far as rectifiers for low currents are concerned. It will be even possible to use pumpless rectifiers already in the near future for higher currents combining rectifiers with only one anode each to form one unit. B.B.C. have also studied the question of the gas filling and their results agree completely with those of Mr. Gerecke. It is possible to eliminate the excess voltages, which occur in cold rectifiers. However, the pressure of the gas should not be too high, else the danger of back fires increases. The danger of the excess voltages occurring in cold rectifiers should not be overestimated. According to the great experience of the B.B.C. it is not necessary to install surge protective devices with all rectifiers.

Fig. 4 gives the load diagram of a pumpless rectifier working in a traction substation and shows that the rectifier is able to carry sudden peak loads. Owing to these good results the firm has started the regular manufacture of pumpless rectifiers. Fig. 5 shows the rectifier L 06 for 500 amps. More than 10 rectifiers of this type have already been sold. The rectifier, L 36 for 1200 amps., is shown in Fig. 6. In Fig. 7 an air-cooled rectifier with pump for 3000 amps., is shown for comparison. It can be seen, that the pump is only an accessory part of the rectifier which will not have a great influence on the total cost. According to B.B.C. the abolishment of the vacuum pump is not of such importance for large rectifiers as for small ones. Furthermore, one has to consider that the inspection of a pumpless rectifier in case of a failure cannot be made on the site owing to the complicated method of degasing. For repairs a pumpless rectifier has to be sent to the factory. A rectifier with a vacuum pump can be repaired on the site, which is an advantage and should not be undervalued. Therefore, the firm has adopted the view that pumpless rectifiers have economical advantages for low outputs. For high outputs, the cost of the pump is only a small fraction of the total costs, and the advantages of a rectifier with a vacuum pump which may be inspected on the site, prevail.

In the further discussion Mr. Gerecke in answer to a question, thought that pumpless rectifiers could be used for voltages of 1500 to 3000 volts, but the development had started with voltages of 600 to 900 volts. To a further question, if by filling with rare gas the danger of back fires would become more imminent, Mr. Gerecke answered: In order to choose the right sort of rare gas, a long series of tests have been made with helium, neon, argon, krypton and xenon. It has been found that the properties of some of these gases come very near to those of the vapour of mercury. Furthermore, the dimension of the grids and the pressure of the rare gas are of great importance.

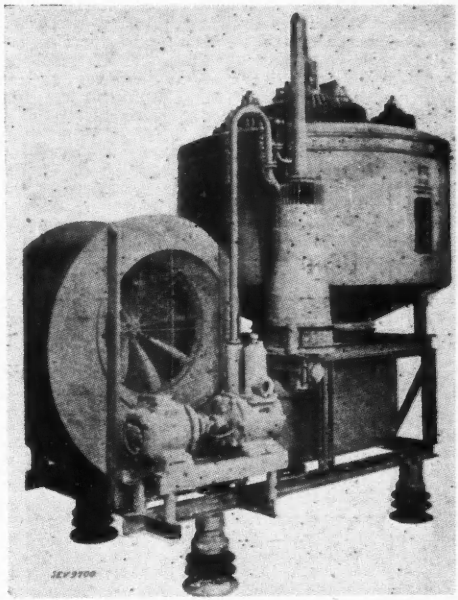


Fig. 7. Air-cooled Rectifier with Vacuum pump of Brown Boveri for 3,000 amp.

ANODIC COATINGS ON ALUMINIUM—THEIR BRITTLENESS

By H. N. Hill and R. B. Mason. (From *Metals and Alloys*, June, 1942, Vol. 15, No. 6, pp. 972–975).

Artificial oxide coatings on aluminium may be of various degrees of hardness, depending on the way they were made. The coatings formed by chemical treatment only, sometimes called dip coatings, are relatively soft and may be even of a powdery nature at times. The oxide coatings produced by electrochemical means (that is, by anodic treatment in various electrolytes) are quite hard and adhere tenaciously to the base metal.

Anodic oxide coatings on aluminium in some respects are comparable in hardness with their mineral counter-

part, corundum. The thick oxide coatings produced in sulphuric or oxalic acid electrolyte by anodic treatment at or below room temperature are hard, abrasion-resistant and corrosion-resistant. The film produced in chromic acid under the usual operating conditions are much thinner than those produced in sulphuric and oxalic acids. These hard oxide films on aluminium are naturally also brittle.

Some tests have been made to measure the brittleness of various types of anodic coatings on 53 S–T aluminium alloy sheet. These tests were made to determine whether or not such coatings are brittle enough to be used in the " brittle coating " method for indicating strain distributions on the surface of a stressed piece. Extreme brittleness of the coating would not be essential to the use of the method for indicating the location of points of maximum stress ; but to be satisfactory for obtaining quantitative results, the oxide coating would have to crack when subjected to strains well within the elastic range of the base metal. The data are also of general interest in connection with the commercial applications of these coatings.

Qualitative Test of Brittleness.

A qualitative observation of brittleness is the fact that a crackling noise can generally be heard when the underlying metal is bent slightly. Fine hairline cracks can be seen with the naked eye, provided the incident light strikes the surface at the proper angle. These extremely fine cracks may almost entirely disappear when the metal is bent back to its original position. When aluminium sheet, coated with a hard oxide film, is cut with shears or scratched with a scriber, fine hairline cracks will develop in the region where the base metal was disturbed. Although the oxide coating is brittle, it adheres tenaciously to the base metal and repeated bendings do not cause it to scale off or loosen.

Although the anodic coatings under consideration are hard and brittle, they are also porous. The pores run from the oxide surface to the metal base and give the coating a fibrous structure. An oxide coating can be sealed by closing the pores in some manner. Sealing is usually accomplished by immersion in boiling water. This treatment converts alumiuium oxide (Al_2O_3) to the monohydrate ($Al_2O_3H_2O$) and seems to change the fibrous nature of the coating, especially in the outer layer. Sealed oxide coatings may have a lower abrasion resistance than unsealed coatings.

Six Special Test Pieces.

For the purpose of evaluating the brittleness of various anodic coatings, six standard tensile specimens were machined from a piece of B. & S. No. 14 gauge (0.064 in.) 53 S–T sheet (Nominal composition : Magnesium 1.3%, silicon 0.7%, chromium 0.25%) which had a tensile yield strength of 36,000 lbs. per sq. in., and an elastic limit of about 22,000 lbs. per sq. in. Since the modulus of elasticity of 53 S–T is about 10,000,000 lbs. per sq. in., the unit strain corresponding to the elastic limit stress of 22,000 lbs. per sq. in. would be about 0.0022 in. per in. Therefore, for the coating to crack within the elastic range of the base metal, cracks would have to occur at some value of unit strain less than 0.0022 in. per in.

These specimens were given oxide coatings by various anodic treatments, as described in Table I. In two cases the oxide coating was sealed.

The specimens were loaded in an Amsler hydraulic testing machine, using Templin grips. Load was applied in increments, so as to produce uniform increments of strain. Strain was measured with a modified Riehle extensometer of 2-in. gauge length, with which strains as small as 0.00002 in. per in. could be estimated. After each increment of strain had been attained, one side of the specimen was examined for cracks, through a 42-power

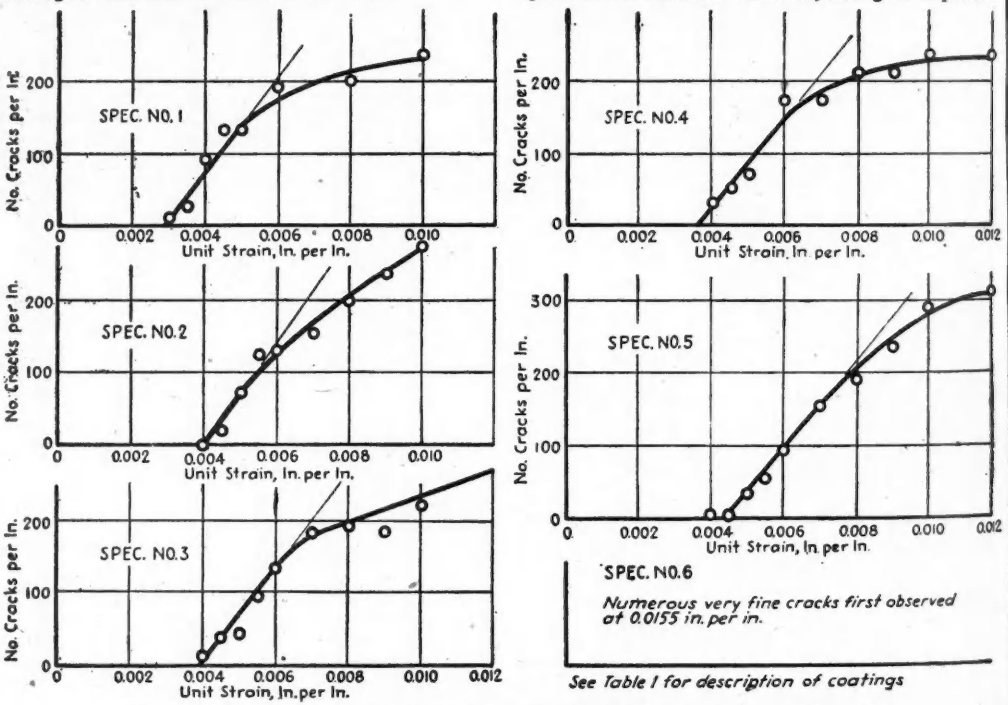

SPEC. NO. 1
SPEC. NO.2
SPEC. NO.3
SPEC. NO.4
SPEC. NO.5
SPEC. NO.6
No. Cracks per In.
Unit Strain, In. per In.
Numerous very fine cracks first observed at 0.0155 in. per in.
See Table I for description of coatings

Fig. 1. Data plotted for the six specimens showing brittleness of the anodic coating.

TABLE I.

Anodic Treatments given 53 S–T Tensile Specimens.

Specimen No.	Electrolyte	Current density, (amp. per sq. ft.)	Temperature (Deg. Fahr.)	E.M.F. (Volts)	Time, (min.)	Sealing Treatment
1	H_2SO_4, 15 %	12	76	17	30	Unsealed
2	H_2SO_4, 15 %	12	76	17	30	30 min. in boiling distilled H_2O
3	H_2SO_4, 15 %	12	76	17	30	30 min. in boiling 5 % $K_2Cr_2O_7$
4	H_2SO_4, 7 %	12	74–83	20	30	Unsealed
5	$(COOH)_2$,3 %	12	80–97	46	30	Unsealed
6	CrO_3 9.5 %	2.5	95	40	30	Unsealed

microscope. Because of the slight stress concentration produced by the change in section, cracks in the coating should occur first at the point of tangency of the fillet and the straight side of the reduced portion of the specimen. One such region was examined continuously as each increment of strain was being applied, and the strain reading was recorded at which the first crack was observed.

The appearance of a crack at this point of tangency was followed by the formation of cracks on the reduced portion of the specimen, at a slight increase in the amount of strain. Further straining increased the number of cracks. One of the first cracks to appear on the reduced portion of the specimen was marked as a reference. As each additional increment of strain was attained, the distance was measured between this reference crack and the fifth crack from it (always in the same direction). This reading gave a measure of the spacing of the cracks. The spacing was not uniform, but the measurement thus obtained provided a reasonable average value.

The data obtained have been plotted in Fig. 1 for the six specimens tested. These curves show the density of cracks, expressed as the number of cracks per inch, plotted against the unit strain. Deviation of the plotted points from the faired curves reflects the limitation of accuracy of the determination of the crack density.

Effect of Stressing.

It was considered possible that cracks which are too small to be readily seen under the microscope might be formed when an anodically coated aluminium specimen is bent or stressed. When an oxide coated specimen of aluminium is placed in a solution containing 20 g. per *l.* of sodium chloride and 1 g. per *l.* of copper nitrate, the ions of the solution penetrate the weak spots; copper is deposited on the aluminium, and readily visible. Using this method of test, it should be possible to detect even minute cracks in the oxide film.

A piece of 53 S–T 14 gauge sheet, 9 in. by ½ in., was anodically coated in 15 % sulphuric acid for 30 minutes, and then sealed in boiling water. The coated specimen was then supported at both ends and loaded so as to produce a stress which was slightly less than required to cause visible cracking of the oxide coating. The specimen was immersed in the test solution while under stress and allowed to stand for about 30 mins. There was no indication of any " cracks " in the oxide coating although the coating had been penetrated by the solution in many spots.

Another similar specimen was bent until definite cracks appeared in the oxide coating. The limiting cracks which could be seen under the microscope were marked with a pencil and the specimen placed in the sodium chloride-copper nitrate solution and left for about 5 minutes. Copper was deposited along the cracks but there was no indication of other cracks in the oxide coating beyond the region of visible cracks. A photograph of the surface with copper deposited along the cracks is shown in Fig. 2. From the results of this test it can be concluded that either this method is not sensitive enough to detect invisible cracks or, as is probably the case, no cracks occur in the oxide coating which are not visible under the microscope.

Table II gives the values of unit strain corresponding to the appearance of the first cracks in the various specimens. Curves of crack density plotted against unit strain are shown in Fig. 1 for all of the specimens tested, with the exception of specimen No. 6.

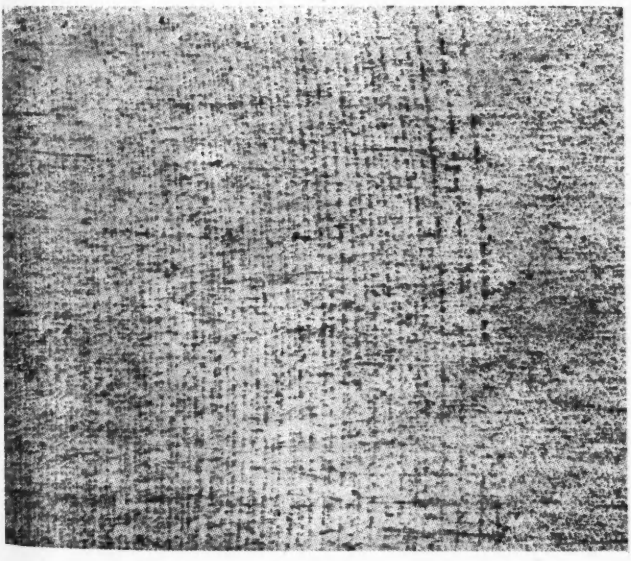


Fig. 2

TABLE II.

Unit Strains at which first cracks appeared in the Straight Portions of the Specimens.

Specimen No.	Unit Strain at First Crack, in. per in.
1	0.0029
2	0.0040
3	0.0039
4	0.0036
5	0.0045
6	0.0155

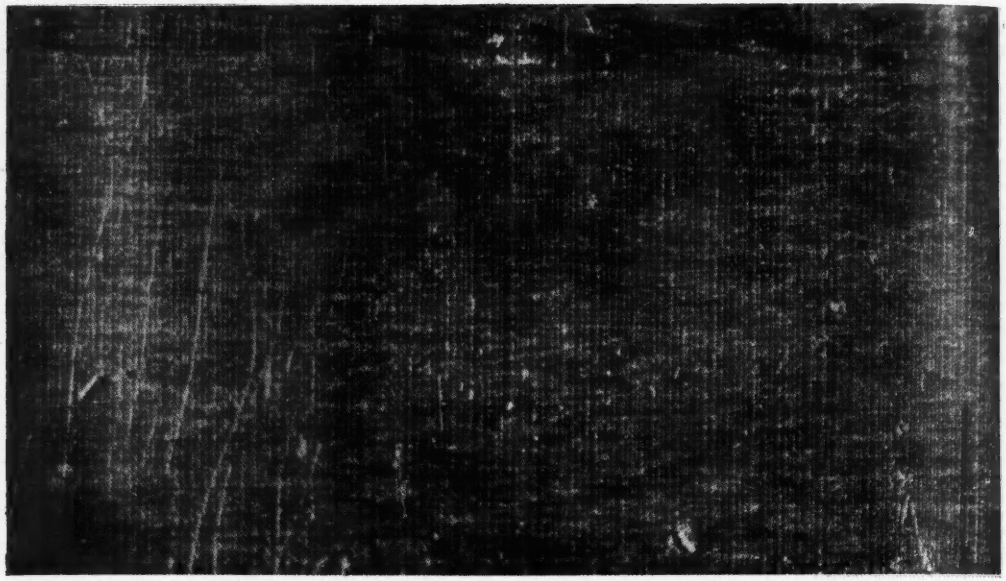


Fig. 3. Cracks in anodic coating on specimen No. 5 after being subjected to a unit strain of 0.012 Mag. about 7x.

No cracks were visible on the coating on this specimen until a unit strain of about 0.0155 in. per in. had been reached, at which time an intricate network of very fine cracks was observed. These cracks were quite different from those which appeared on the other specimens. The photograph shown in Fig. 3 is typical of all of the specimens, except No. 6. The cracks produced a series of parallel lines running continuously across the specimen.

The general similarity of the curves in Fig. 2 indicates that the coatings produced by the different treatments behaved in about the same manner. After the first crack appeared, the density of the cracks increased at a constant rate with an increase in strain up to a certain point, beyond which the number of cracks increased at a decreasing rate.

Conclusions.

In conclusion, it may be stated: The oxide coatings produced on aluminium alloy 53 S–T by anodic treatment were not brittle enough to be satisfactorily used to indicate the strain distribution on the surface of a stressed specimen. In fact, cracks did not occur in the coating until the material had been strained beyond its elastic limit (0.0022 in. per in.). Sealing by immersion in boiled distilled water or a hot dichromate solution slightly reduced the brittleness of the coating. The data show that these coatings are surprisingly elastic in spite of their mineral-like hardness.

INCREASING WELDING EFFICIENCY

By Dipl. Ing. HERBERT SCHULZ. (From *Zeitschrift des VDI.* Vol. 86, Nos. 23/24, June 1942, pp. 369–371).

FOR maximum welding efficiency oxy-acetylene gas has proved to give best results not only for welding steel but also for light metals and zinc, lead, etc. Its flame temperature and specific flame efficiency are both superior to those of other welding gases.

Considerable attention should however be given to the problem of gas supply; whenever welding is performed at several places in a workshop, the most economical solution is to arrange for a central supply station and, of course, the necessary tubing. With this solution uninterrupted welding is possible as it does not require the attention of the welder but of a separate attendant who, incidentally, is less skilled than the welder.

The acetylene is supplied by a generator or from steel drums; the oxygen from steel drums or from a supply of liquid oxygen. The capacity of the supply station must be calculated according to the maximum demand at any time and should be carefully estimated. The capacity of the acetylene generators is determined by

(1) the continuous output which is approximately half the maximum output stated on the generator, and

(2) the capacity of the carbide container. The diameter of the tubing to the welding stations can be easily calculated from $D = k.Q/v.$, where Q is the quantity of flow in the tube, v. the velocity of flow and k. a coefficient according to the units chosen. A small allowance must be also made for frictional losses at the bends, etc.

Hand Welding.

Of considerable importance is the discrimination of the work into welding " to-the-right " and " to-the-left".

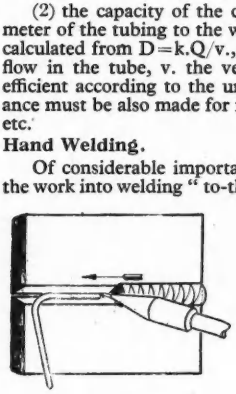


Fig. 1

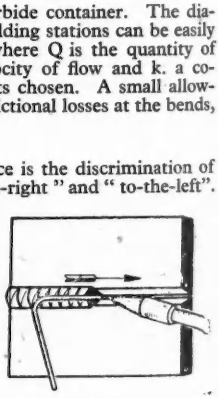


Fig. 2

Welding "to-the-left" is the method of applying the flame between wire and seam, "to-the-right" when the wire is applied between flame and seam. If the working material is steel, up to 3 mm. sheet thickness the "to-the-left" welding is used, Fig. 1, since with the "to-the-right" method, holes might be burnt into the metal.

For sheet thicknesses above 3 mm., however, the "to-the-right" method should be used, Fig. 2, as this is much quicker as can be seen from curves (a) and (b) in Fig. 3. Moreover, the "to-the-right" welding is very efficient inasmuch as it does not drive the molten metal in front of the flame and thus cold spots in the seam do not occur.

In view of the advantages of the "to-the-right" method of welding it is a remarkable fact that so far this method is not utilised to full capacity. The reason for this neglect in workshops is probably due to the fact that welders are still inexperienced with this method and consequently when tried the result is not satisfactory. To overcome this difficulty, works managers should focus sufficient attention to this problem and re-train welders to the new technique. As the saving in welding efficiency is about 30%, the importance of this method cannot be over stressed.

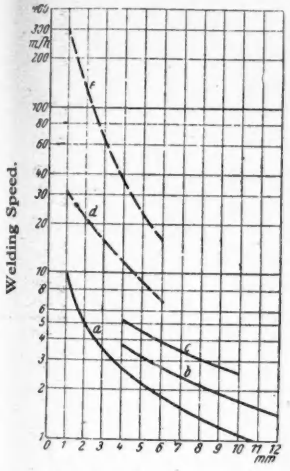


Fig. 3. Sheet Thickness.

Another possibility of increasing the welding efficiency is by using two-flame welding. This method is adaptable to horizontal welding of all types of sheet thicknesses between 3 and 10 mms. The welding speed of two-flame welding is shown by curve (c) in Fig. 3.

The difficulties experienced with the introduction of the "to-the-right" welding method become still more drastic with two-flame welding. This is then the reason for many two-flame welding apparatuses lying idle in workshops. Again, the solution would be proper training of the welders.

Mention should be made of a special welding method; two-side welding, which is performed by two welders simultaneously. This method is practicable to sheet thicknesses above 8 mms., and welding is performed on vertical work. Although the welding speed of this method lies between the "to-the-right" and "to-the-left" method, its adoption is often justified by the saving of a considerable quantity of heat. As an example the welding of a 10 mm. V. seam would require about 2500 l/h oxy-acetylene mixture while only half that much is used when 2-side welding is applied.

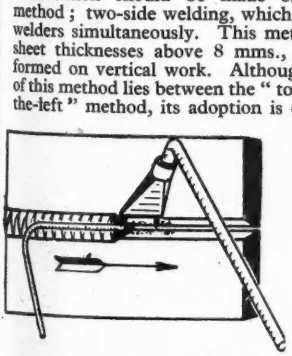


Fig. 4. Two-flame welding.

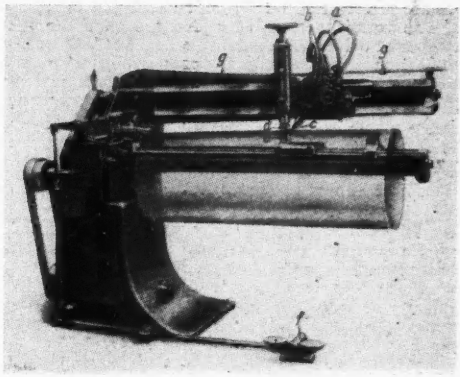


Fig. 5

Machine Welding.

The change over from hand to machine welding is, as would be expected, accompanied by a considerable increase in welding speed. Welding machines designed for sheet thicknesses from 0.5 to 5 mm. have given excellent results in practice. For thinner sections the machines are much simpler than those for heavier work where an oscillating motion is required for the wire and burner. This technical requirement forms the greatest obstacle, so far, to the design of an all-round satisfactory welding machine for heavy gauge sheet.

As mentioned before for sheet thicknesses up to 5 mms. simple machines are needed, and in practice two kinds have been accepted. With one type the burner is automatically guided over the fixed work, while in the other type the burner is fixed and the work is moving. Fig. 5 shows a machine of the first type. The burner is usually of the water-cooled, two-flame type. It is travelling along the upper arm of the machine at a speed corresponding to the sheet thickness. The speed can be steplessly adjusted. The lower arm of the machine holds the work by means of a clamping device. The efficiency of such machines is an improvement on the 2-flame hand operated welding (see curve (d), Fig. 3).

Fig. 6 shows a two-way welding machine of the second type, that is, with fixed burner. These machines are especially suited for tube welding up to 140 mm. diameter and 6 mm wall thickness. The principle of operation of the machines is as follows:

The sheet metal is rolled to a tubular form in a special machine and is then introduced into the welder through a pair of driving rollers (a). The work is then guided by a pair of guiding rollers (b), under the flame and the

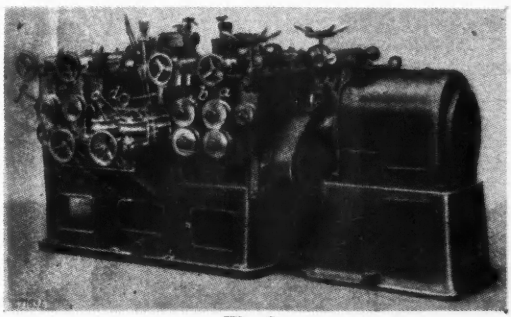


Fig. 6

edges are pressed together by yet another pair of rollers (c). The work is then passed from the machine through a number of driving and guiding rollers.

These machines are usually made fully automatic controlling the various phases of operation, such as rolling, milling, welding and cutting. The welding burners are of the multi-flame type, the number of flames varying between wide limits.

The welding speed and hence, the output of such machines is very large; it depends to a large extent upon the burner size. Fig. 3 shows in curve (e) the speed of an automatic tube welding machine of 6 to 10 flames.

NEW TYPE SINGLE JET CARBURETTOR

By Dr. Ing. Muller. (From ATZ, *Automobiltechnische Zeitschrift*, Vol. 21, No. 10, November, 1941, pp. 537–539.)

The "Diflex" carburettor, manufactured by the Praga Works, represents a novelty in carburettor design. This carburettor opeates with a single jet at all speeds, and a choke tube which has an adjustable cross-section to suit the whole of the speed range. For starting the engine a cold starting valve is provided. The fuel level in the float chamber, 1 (Figs. 1–3) is kept at a constant height by the valve needle. The fuel enters the jet tube, 8, through the channels 2, 3, 4, 5 and 6 and through the set, 7. The channel, 5, has a calibrated bore, corresponding to the size of the engine, and regulates the fuel supply to the jet. The newest experimental models have a spring loaded needle valve provided to regulate the cross-section of the channel, 5, at partial loads. This arrangement is not shown in the figures, but tests indicate that it is an effective means to reduce fuel consumption.

The jet tube is, of course, located at the narrowest section of the atomizer. The air is admitted, as indicated by the arrows, through fine holes in the tube, 8 (Figs. 2 and 3). The tube is fixed by the nut, 9, and prevented from rotating by the pin, 10.

At the air inlet of the mixing chamber a filter is provided, and a valve controlled by a bowden cable, for cold starting. Valves, 11, are forced by the springs, 12, against the tube, 8, when the engine is stopped (Fig. 3). On opening the throttle valve, the valves, 11, take up a position corresponding to the engine speed. The stops, 13 (Fig. 3), limit the operation of valves, 11. The crooked ends of the atomizer valve are guided by plates, c (Fig. 3), small perforations are made in the atomizer valve to equalize the pressure.

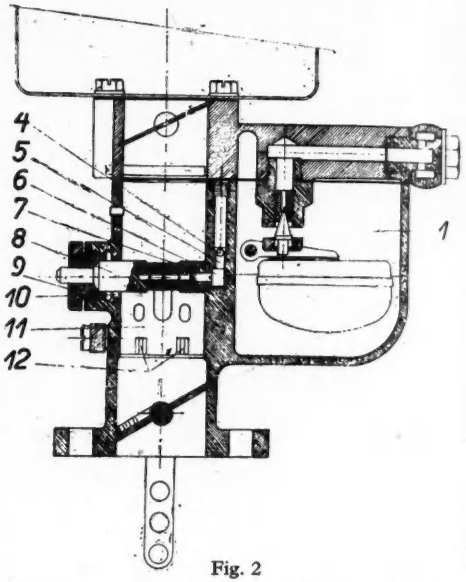


Fig. 2

Regulation and Operation of Diflex Carburettors.

If the engine is started from cold, the air valve is completely closed, the engine is then speeded up by depressing the accelerator. If the engine starts drawing in a mixture, the valve is opened accordingly; if the engine stops, the valve was opened too wide. If, on the other hand, the engine speed is not increased when the accelerator is depressed, the valve needs further opening as the mixture must be made poorer. At low ambient temperature, the engine is run for a short time with the valve not fully opened, so as to heat up the engine. As soon as the load is applied, the valve is fully opened.

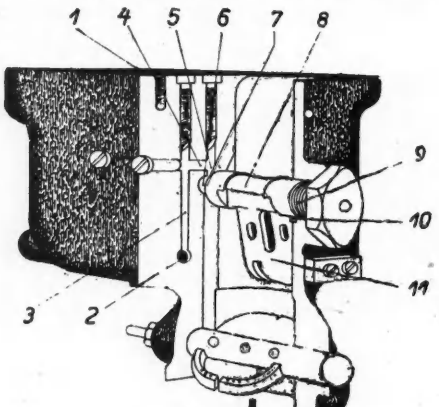


Fig. 1

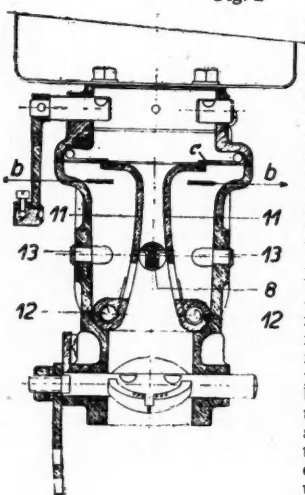


Fig. 3

In spite of the very simple construction, the proper functioning of Diflex-carburettors depends upon the uniform and frictionless operation of the atomizer valves. In the appropriate opening of the valves, corresponding to the engine speed, the springs, 12, play an important part. These, as well as the stops, 13, are adjusted to suit the engine type.

As mentioned before, the float level regulation takes place inside the float chamber and can be checked after the cover is removed. To check the operation of the atomizer valves, the air filter has to be removed, on pressing down the accelerator, the valves should smoothly and uniformly lift from the tube. If this is not the case the spring tension must be altered or the valve attachment loosened.

Diflex carburettors were fitted to various types of engines for experimental purposes. As an example, a "Praga" RV-3, 5–1 motorcar is quoted which had originally a Solex-carburettor, type 40 RIFP.

Car-burettor.	Light-load Jet.	Main Jet.	Ato-mizer.	Channel. "5."
Diflex ..	—	135	45	1.7
Solex ..	55	155	31	—

No regulating valve in channel, 5, was fitted to the carburettor.

Power and consumption figures were established for both types of carburettors under normal running conditions. The top speed, corresponding to the throttle valve adjustment, was 2,400 r.p.m., although the engine was actually capable to run at 3,500 r.p.m. The engine had six cylinders and Ricardo type combustion chambers. The power and torque curves are shown in Fig. 4, and it can be seen that, especially at top speed, they are higher for the Diflex carburettor. The corresponding fuel consumption curves are shown in the lower half of

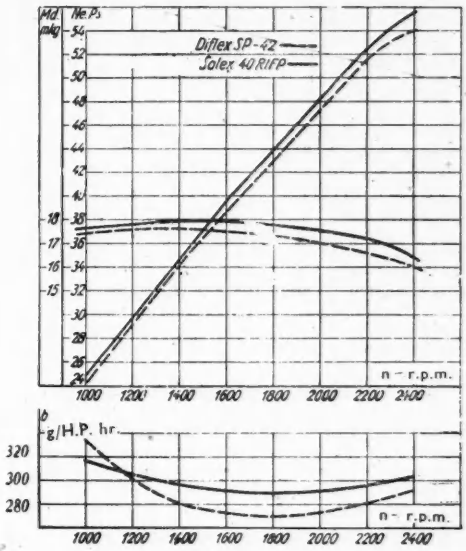


Fig. 4

the graph, and again fall out favourably for the Diflex carburettor. An interesting feature noted was the excellent acceleration under full load between 600–2,400 r.p.m., being 5.5 seconds, against 7.5 seconds, for the Solex carburettor.

A number of tests were also made in actual operation. The route included mountain path, as well as town roads, and the fuel consumption observed was in each case considered as very favourable. Further development and utilization of Diflex carburettors is therefore indicated.

CARBONISATION PROPERTIES OF DIESEL FUELS

By F. KNEULE. (From *Zeitschrift des VDI*, Vol. 86, Nos. 5/6, February, 1942, pp. 76–78).

PARALLEL to the development of Diesel engines, new types of fuels were also introduced which, though satisfactory from the combustion point of view, entail complications by forming carbon residue adhering to the interior of the combustion chamber.

Deposition of tar, soot, and unburnt fuel particles on the walls of the combustion chamber can be easily prevented by raising the wall temperature or by increasing the proportion of excess of air. Carbonisation of the nozzle orifice and nozzle valve, however, constitutes a more serious problem because of its ill-disposed consequences and difficulty of prevention.

Process of Nozzle Carbonisation.

The fuel leaving the nozzle orifice at the end of the injection period possesses a small kinetic energy only, as, at this stage, the injection pressure is low and the pressure in the cylinder high. Dribbling is, therefore not an unusual phenomenon, especially if wear or high temperature causes leakage through the valve.

If the nozzle body attains a certain temperature, fuel will not only evaporate from the nozzle orifice, but cracking will also take place, and the adhesive force between metal and carbon produced by cracking will suffice to start carbonisation. The seriousness of carbonisation will depend upon the nature of the residue; brittle or flaky residue will cause temporary disturbances only for a longer period, before sufficient residue has accumulated on the nozzle to reduce appreciably the engine power.

At still higher temperatures, cracking can also occur in the clearance between valve and nozzle body, thus reducing the free play of the valve and eventually stopping valve operation altogether.

Influence of Engine Design.

As temperature and nozzle construction are the primary factors affecting carbonisation, the means of preventing this must be by appropriate nozzle design and temperature control. The nozzle design should, therefore, embrace such considerations, by preventing fuel from remaining between valve and valve seat during the "closed" period. A rigid, symmetrical nozzle body will usually give satisfactory results as it will not deform under the action of heat. An elastic fitting of valve on the valve seat and accurate guiding of valve motion will also promote tight closing. Flat valve seats have an advantage over conical ones, inasmuch as they prevent leakage even in the case of the valve stem not being accurately centered. With multi-hole nozzles a permanently tight sealing is more difficult to obtain, as the small holes in the closed fuel piping may allow high pressures to prevail.

The nozzle body should be snugly enclosed by a well

cooled material; the nozzle temperature lies between 150 and 180° C., but can reach 350° C. under unfavourable conditions. If cooling is effective, however, the residue formed will be loose and constantly removed by the "blast" of the injected fuel spray. Direct cooling, applied to the nozzle, has been designed by many manufacturers, but this can be employed only in very large engines.

Influence of Fuel.

Practical experience has revealed that there are certain fuel characteristics which affect carbonisation unfavourably. These are: High tar content, high proportion of heavier fuel fractions, high self ignition temperature and high viscosity. Non-carbonising fuels are used, sometimes, as dope to improve inferior fuel sorts. In this respect Heinze and Marder (*Ol und Kohle*, Vol. 37, 1941) found that even small quantities of dopes, used to lower the self ignition temperature, are liable to increase the proneness to carbonisation of the fuel.

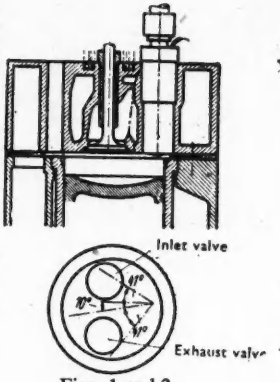
Laboratory Testing.

Various laboratory test methods have been developed to determine carbonisation properties of fuels, by evaporating the lighter fuel fractions and estimating the percentage weight of the residue. The methods were in part similar to those used for lubricating oil testing, and in part developed to suit the particular purpose of fuel testing. A standardisation of diesel fuels, on this basis, did not follow as yet, since the behaviour of fuels in engines does not correspond to the expectations based on the tests. It has been found necessary, therefore, to investigate whether suitable laboratory tests can be devised at all, and if so, what are the requirements for practical adaption.

Engine Testing.

The choice of a suitable parameter for measuring carbonisation properties entailed some difficulties at the beginning. Finally, measurement of power, i.e. drop of power, was found to give a suitable scale, since carbonisation of the nozzle produces a disturbance of the fuel spray, thus deteriorating mixing which is finally manifested in reduction of power. The nature of the deposited carbon is also gauged by this method, inasmuch as the decrease in power will be the more rapid the tougher the deposit.

In order to shorten the test period, carbonisation was speeded up by heating applied to the nozzle, (Figs. 1–3). The engine used for the test purposes was a single cylinder, open chamber, vertical engine which was run on the fuel to be tested until the power has been decreased by 25%. The time taken for this was observed and used as a scale for comparative purposes.



Figs. 1 and 2

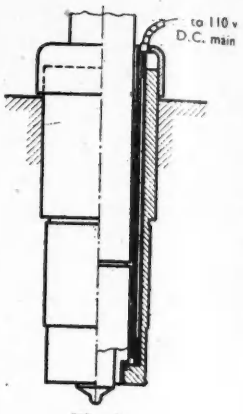
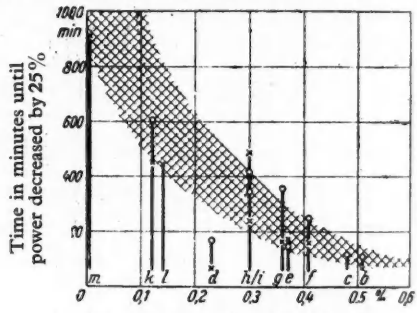


Fig. 3

Experimental Results.

Table I comprises the most important chemico-physical characteristics of the fuels tested, as well as the corresponding Cetene numbers. The last two columns give the test times for two different nozzle temperatures.

Unfortunately the results do not indicate a universally valid relationship between fuel characteristics and carbonisation properties in engines. In Fig. 4, the test times are shown plotted against values obtained from



Test results according to Conradson.

Fig. 4

TABLE I.

Fuel	Density at 20° g/cm³.	Distillation °C.	Viscosity at 20° °E.	Self Ignition Temp. °C.	Constituents Aromatic %	Constituents Unsaturated %	Cetane No.	Asphalt Test %	Conradson's Test %	Engine Tests 205°C. min.	Engine Tests 225°C. min.
a	0.873	318	2.90	92	14	41	62	0615	1.89	111	83
b	0.847	293	1.62	96	2	19	70	0,10	0.51	110	70
c	0.846	295	1.61	90	5	15	59	0,107	0.48	115	93
d	0.891	238	1.21	86	4	53	35	0,049	0.23	170	70
e	0.904	272	1.49	87	6	53	37	0,177	0.37	148	174
f	0.922	306	2.02	75	together	72	37	0,148	0.41	250	160
g	0.906	300	1.94	93	9.5	25.5	33	0,071	0.31	357	168
h	0.892	235	1.24	84	5	53	36	0,023	0.30	415	240
i	0.899	275	1.46	77	7	56	37	0,036	0.30	344	490
k	0.899	267	1.39	75	6	58	53	0,024	0.12	606	450
l	0.890	273	1.39	86	7.5	48.5	44	0,009	0.14	600†	445
m	0.851	252	1.19	71	1.5	30.5	65	0,032	0,003	600†	913

† Experiments were broken off after 10 hours, when no decrease in power was yet observed.

Conradson's test, and, considering the difficulties of the test method, the two sets of results seem to be in satisfactory alignment. As was expected, with higher nozzle temperature the test times were much shorter. Fuels (e) and (i) show some discrepancy, but this is probably due to inaccuracy of measurements.

Quite out of proportion is the results with fuel (d) which was produced by cracking. This fuel reveals such a strong tendency to carbonisation which could not be expected from the Conradson's test, and as the results in actual engine tests were similar, it appears that for cracked fuels the Conradson's test is not conclusive.

Summary.

Although no final conclusion can be suggested on the basis of the results, nevertheless it appears that the relatively simple Conradson's test can be used, in most cases, to predict the behaviour of fuels as regards carbonisation. As this fact is strongly supported also by practice, the DVM Association has suggested this method to be adopted for standard measurement of carbonisation properties of fuels.

The results will not be, of course, of universal validity for all types of engines and nozzles, but as the test conditions are most unfavourable (i.e. carbonisation will be the strongest) a general indication will be given as to their behaviour in engines.

The future aim of research work in this line should include testing of all types of engines in commercial use and to determine whether or no cracked fuels in general have a different character. In the latter case it would be possible, by applying a multiplying factor, to apply the Conradson test also to cracked fuels.

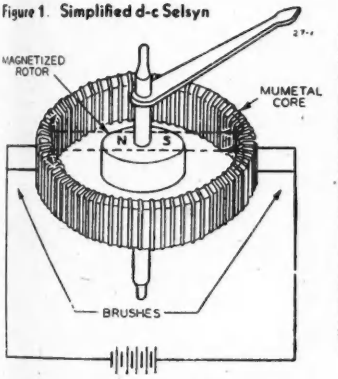
A D.C. TELEMETER OR D.C. SELSYN FOR AIRCRAFT

By R. G. JEWELL and H. T. FAUS, General Electric Co., Lynn, Mass. (From *Electrical Engineering*, vol. 61, June, 1942, Transactions, pp. 314/17.)

THE large number of indications, which must be transmitted accurately to the instrument board of an airplane has created a demand for a telemetering system of small size and light weight. For many years the A.C. Selsyn telemetering system has given excellent service in marine applications. Variations of this system are used successfully in aircraft. A search for a simpler and lighter system which will operate on direct current, has led to the development of D.C. Selsyn telemetering system.

A simplified concept of the operation of the D.C. Selsyn may be obtained by referring to Fig. 1, which shows a coil wound on a toroidal iron core. Two brushes supply direct current at diametrically opposite points of the continuous single layer winding. A magnetic field is set up inside the core and follows the brushes as they are rotated. A polarized permanent magnet placed inside the core revolves so as to keep its direction of magnetization in line with the brushes. Fig. 2 shows the circular rheostat transmitter tapped at intervals of 120 degrees. These taps are connected to similarly spaced taps on the receiver. The permanent magnet rotor of the receiver will turn to a position having the same relation to the taps on the receiver

Figure 1. Simplified d-c Selsyn



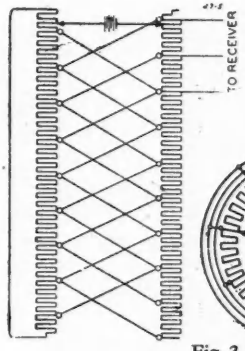


Fig. 2. D.C. telemetering System,

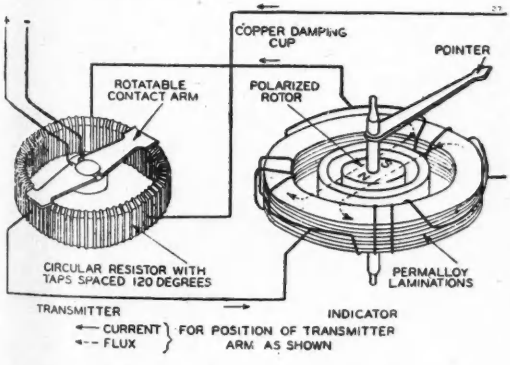


Fig. 5. (Left) Linear transmitter for D.C. Selsyn. Indicator makes three revolutions as brushes move from end to end of transmitter.

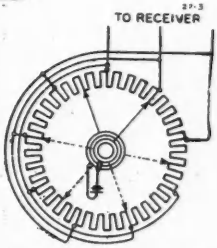


Fig. 3. D.C. Selsyn transmitter. Indicator makes three revolutions per revolution of transmitter.

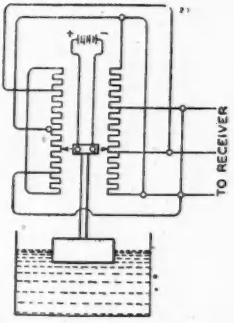


Fig. 4. (Right, above) Linear transmitter for D.C. Selsyn. Indicator makes one revolution as brushes move from end to end of transmitter.

winding as the brushes have to the corresponding taps on the transmitter winding. Connections similar to other polyphase windings may also be used and the windings may be concentrated in coils rather than being uniformly distributed on the core.

The permanent magnet of high coercive force in combination with a copper damping shell fixed in the air gap between the rotor and the stator of the receiver gives effective damping with high torque. The high torque obtained with light weight is one of the outstanding advantages of this system. The light moving element requires neither ball bearings nor jewel bearings, since friction errors are inappreciable with steel and bronze bearings. Since these bearings are lightly loaded, they withstand severe vibration very well. The brushes of precious metal alloys operating at carefully determined pressures give a reliable contact with very little friction. Life tests have shown these brushes to be capable of operating for over 40,000,000 complete revolutions of the transmitter.

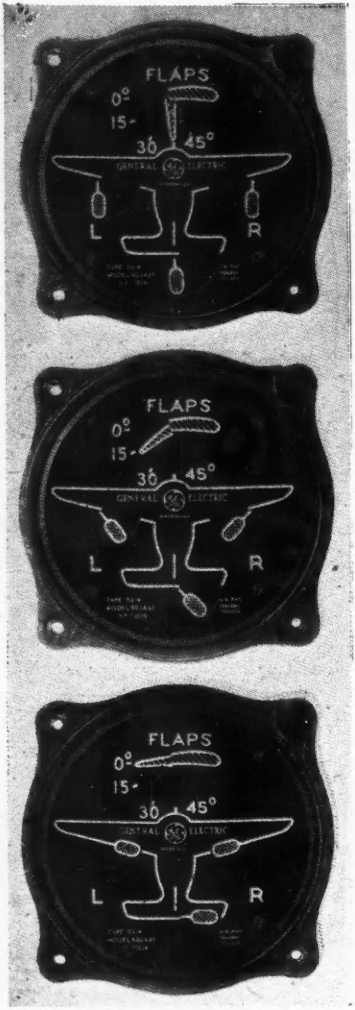


Fig. 7. D.C. Selsyn position indicators showing three positions of flaps and landing gear.

In the transmitter shown in Fig. 2 the ratio of angular motions of the transmitter and the indicator is unity. By making the number of evenly spaced taps equal 3n, where n is the number of indicator revolutions per transmitter revolution, it is possible to obtain a ratio equal to any whole number; 2n brushes, 360/2n angular degrees apart, may be used, but the transmitter is operable with only two brushes. Fig. 3 shows a transmitter designed for a ratio of 3.

Fig. 6. D.C. Selsyn position transmitter.

The transmitter shown in Fig. 4 is designed to transmit linear motion without converting it to circular motion by mechanical means. Motion of the brushes

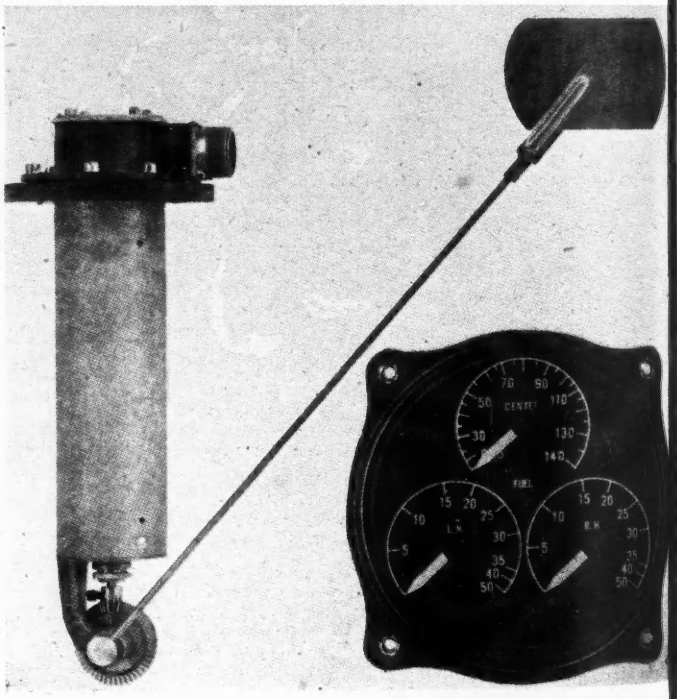


Fig. 8. Typical liquid-level transmitter.

Fig. 9. Typical three-element liquid-level indicator.

for the complete length of this transmitter gives one revolution of the indicator. A multi-revolution modification of a linear transmitter is shown in Fig. 5 (three revolutions for the complete length of the transmitter).

Two or more indicators may be operated in parallel in connection with one transmitter. When the power is disconnected, the indicator will remain unchanged in position. If an indication of power failure is desired, the indicator is provided with a small fixed permanent magnet, which attracts the rotor magnet to a position, at which the pointer is off the calibrated part of the scale. Due to the high torque of the instrument it is possible to use a fixed magnet strong enough to provide a reliable indication of power failure without having any significant effect on the accuracy of the system.

Each indicator element weighs 25 grammes and can be mounted in a 1¼ inch diameter circle. The power required by the transmitter and indicator combination is 1.8 watts, with two indicators connected to one transmitter the power required is 2.2 watts.

Varying the voltage of the supply does not affect the indication provided no power failure indicator is used. With such an indicator a variation of ten per cent. in the voltage will cause a maximum error of one degree. Temperature effect on indications is negligible. An indicator connected to the transmitter will not change its calibration when another indicator is placed in parallel with it. The peculiar shape of the voltage wave does prevent the indicator from following the transmitter linearly. However, this inherent deviation has been determined to have a maximum of 1.3 degrees.

Applications.

A D.C. Selsyn position transmitter is shown in Fig. 6. One of the first applications of the system was for the remote indication of the positions of the landing gear and wing flaps on an airplane. For this purpose four elements were mounted in a single case (Fig. 7), one element being used for each of the three landing wheels and one for the flaps.

The remote indication of oil and fuel pressure is accomplished by connecting a bellows to the oil or fuel pressure line and utilizing the motion of the bellows to rotate the transmitter brushes.

The temperature transmitter has a bimetal helix which rotates the transmitter brushes as the temperature changes.

The manifold pressure transmitter has a partially evacuated sealed bellows in a pressure tight compartment. The manifold pressure is applied to the outside of the bellows, motion of which is transmitted mechanically to the Selsyn transmitter.

The liquid level measurement utilizes the motion of a float to actuate the transmitter brushes (Fig. 8). Magnetic coupling is used between the transmitter outside the tank and the float mechanism inside the tank so as to provide a positive gasoline seal.

PROGRESS IN STRUCTURAL DESIGN THROUGH STRAIN-GAUGE TECHNIQUE

By C. R. STRANG, Douglas Aircraft Co., Inc. (From *S.A.E. Journal*, Vol. 50, No. 8, pp. 346–357, August, 1942.)

THE very words " strain gauge " infer a primary concern with theory, due largely to their past limitation to laboratory work. They are, however, here discussed as a work-a-day aid to the structural designer, substantiating past decisions and supplying information for better future decisions.

Equipment.

I will limit myself almost entirely to the application of electrical resistance strain-gauge apparatus to the solution of typical structural design problems in order to make clear the vastly accelerated progress made possible by the use of this equipment. The resistance units themselves have been described by de Forest and Leaderman in NACA Technical Note No. 744, and a description of the associated amplifying and recording apparatus as developed by the Douglas Aircraft Co., was presented by Messrs. Root, Dickinson and Ostergren in a paper which was presented before the Institute of Aeronautical Sciences, January 27, 1942.

In general, the equipment consists of resistance units of fine wire cemented to the stressed surface under investigation and connected in a suitable bridge circuit. The cross-section, and hence the resistance of the gauge unit, is varied by the elongations or contractions to which it is subjected. The resulting unbalance of the bridge circuit is amplified by electronic means, and recorded by equipment suitable to the nature of the tests; a recording oscillograph is used if the tests are dynamic in nature, or stylus recorders, or even simple visual meter readings are used if the stresses involved are static.

Much of the substantiation of existing theory has been due to the use of mechanical strain gauges. For the most part, such gauges require a highly developed technique of their own. Their nature is such that good results with them are usually only obtained with highly trained personnel and under favourable laboratory conditions. In actual aircraft work, the odd sections to which gauges must be attached, the lack of room for their installation, the rapid variation of stresses in flight or dynamic test work, and the importance of stresses in thin sheet working in the wave state, to which suitable attachment of a mechanical gauge is very difficult, have all resulted in limited use of such gauges. By contrast, the electrical gauge units are themselves small and suffer from none of the foregoing disadvantages. In addition, when they are used with suitable associated apparatus, they are remote reading and self-recording. For each type of stress, there is a special most-suitable use of the gauge units. These special applications for each type of stress will first be considered.

Axial Stresses.

In all work with these gauges, it is necessary to remember that a strain and not a stress is being measured. Therefore, Poisson's ratio corrections must be made wherever stresses not parallel to the axis of the gauge exist. Where simple axial loads with a uniform stress distribution across a cross-section of a member could be assumed to exist, a single gauge would be adequate to determine that stress. Such a fortunate state of affairs is difficult to achieve in the laboratory, let alone in actual aircraft structures and, for that reason, a much more satisfactory procedure is to use a number of gauges so interconnected that an average of the stresses imposed on all of the gauges is read. By this method, a true indication of the average axial stress and of the true axial load in the member may be obtained.

Assuming purely axial stresses parallel to the axis of the gauges, the stress will of course be

$$\text{Stress} = E \times \text{Strain}.$$

Fig. 1 is a comparison of a stress-strain curve

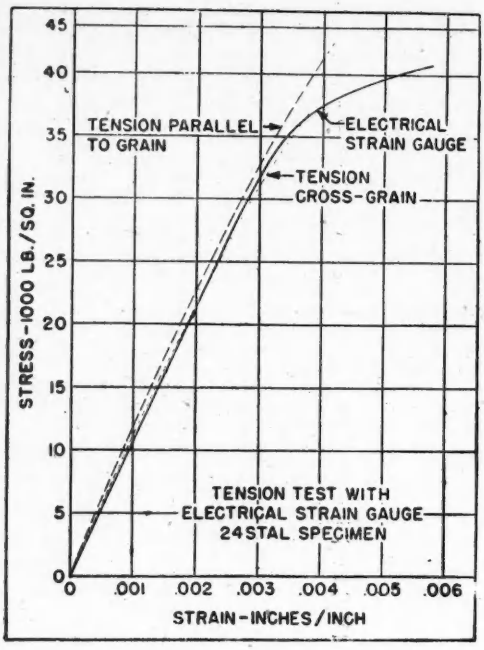


Fig. 1

indicated by electrical gauge equipment with curves determined by more customary means. Note that, at stresses above 37,000 lb./in.², the electrical gauges indicate a much larger increase in strain than should be expected. The useful range of the gauges is, therefore, taken to be of the order of 0.0037 in./in. strain. Whence:

Gauge limit for dural = 10,000,000 × 0.0037 = 37,000 lb./in.²

Gauge limit for steel = 29,000,000 × 0.0037 = 107,000 lb./in.²

These figures indicate that the gauges are suitable for proof tests and flight-test work in which yield stresses are not exceeded for 24 STAL dural or equivalent, and for steel up to ultimate tensile strengths of the order of 125,000 lb./in.²

For flight-test work or static proof tests on aluminium alloys the indicated range is satisfactory. The permissible upper limit is unfortunately not high enough to permit direct simple use beyond the yield stresses in destruction test work. However, the gauge unit's range of usefulness can sometimes be extended by special circumstances.

Bending Stresses.

Where a varying stress distribution is known to exist, and it is desired to determine that distribution, a number of individual gauges located at critical points may be used. From this data, a plot of the stress distribution across the section may be made, and bending and direct stress may be isolated subsequently.

One such case is illustrated by the arrangement used to determine the stress distribution across the unsymmetrical I-section of an extremely large, forged fitting, part of which is shown by Fig. 2. Pickups were distributed as shown, together with a section of the record taken during a landing, by Fig. 3. For con-

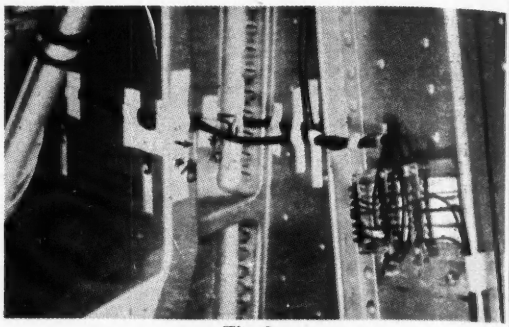


Fig. 2

venience, these circuits are usually so arranged that tensile stresses are indicated by an upward swing of the trace and vice versa for compression. The vertical lines are time lines, 0.10 sec. time interval being indicated by each space. The redistribution of bending stress due to beam curvature is another field that has been investigated by a similar installation.

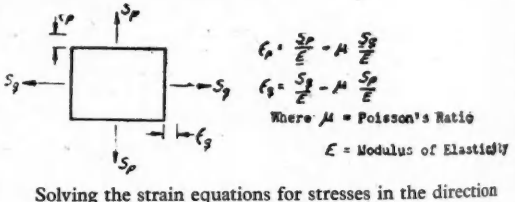
When bending stresses are the sole concern, it is possible to use two gauges on opposite sides of a bending member, interconnected in a single circuit so that the total stress differential (a combination of the tension on one side, plus the compression on the other side) is indicated. Such applications are particularly accurate because the nature of the circuit is such that the two gauges are used on opposite legs of the bridge.

Shear Stresses.

In the general case, with the direction and magnitude of the principal stresses unknown, it is always possible to determine the complete state of stress on a surface if a sufficient number of strains directionally disposed at sufficiently large angles to each other are known.

Literature covering the mathematics of the determination of the state of stress from such data is fairly complete, having been developed originally for use with mechanical gauges. Of all the available methods, the use of three gauges, each forming an angle of 60 degrees with the other, in the shape of a delta, shown by Fig. 4, lends itself most easily to rapid mathematical reduction. The result sought is the determination of the direction and magnitude of the principal stresses from which any desired components, such as the shearing stresses may be established. A sketchy development of the mathematics involved is as follows :—

An element oriented in such a manner that only principal stresses exist is considered. The strain in each direction follows :—



Solving the strain equations for stresses in the direction of the strains, the following fundamental equations are obtained :—

$$Sp = \frac{E}{1-\mu^2} (\epsilon_p + \mu \epsilon_q)$$

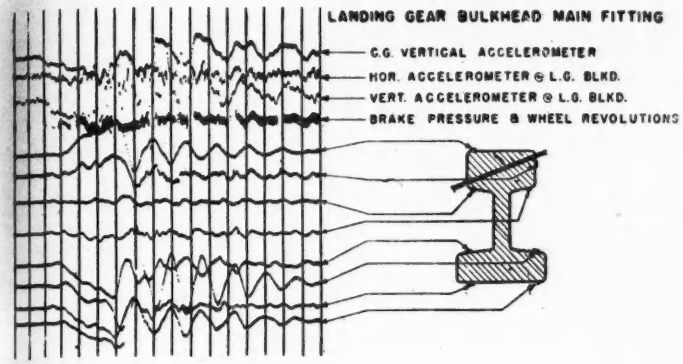
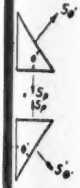

LANDING GEAR BULKHEAD MAIN FITTING
C.G. VERTICAL ACCELEROMETER
HOR. ACCELEROMETER @ L.G. BLKD.
VERT. ACCELEROMETER @ L.G. BLKD.
BRAKE PRESSURE & WHEEL REVOLUTIONS

Fig. 3

Therefore, if the strains are known in the direction of the principal stresses, a solution for the principal stresses can be obtained.

In most cases, however, the direction of the principal stresses is not known. The use of three known strains in a direction 60 degrees to each other will give all of the principal stresses. The direction in relation to any reference line of these stresses may also be determined.

The stress in any plane is :—



$$S_{\theta'} = S_p \cos^2\theta' + S_q \sin^2\theta'$$

$$S_{\theta'} = S_p \sin^2\theta' + S_q \cos^2\theta'$$

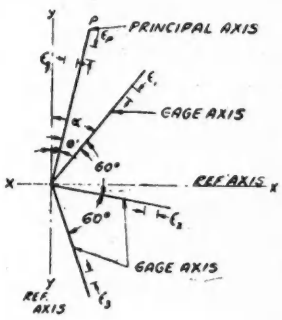
By substituting proper strain values in the above equations, the strain is:

$$\epsilon_{\theta'} = \left(\frac{\epsilon_p+\epsilon_q}{2}\right) + \left(\frac{\epsilon_p-\epsilon_q}{2}\right)\cos 2\theta'$$

the terms $\frac{\epsilon_p+\epsilon_q}{2}$ and $\frac{\epsilon_p-\epsilon_q}{2}$ are defined as ϵ and Δ respectfvaly,

the strain in any direction in terms of the principal strain values

$$\epsilon_{\theta'} = \epsilon + \Delta \cos 2\theta'$$



hown by the figure, the following

ations result:

$$\epsilon_1 = \epsilon + \Delta \cos 2\theta'$$

$$\epsilon_2 = \epsilon + \Delta \cos 2(\theta'+60°)$$

$$\epsilon_3 = \epsilon + \Delta \cos 2(\theta'+120°)$$

angle α is known from the allation of the gauges. A ion of the above equations the following results:

$$\epsilon = \frac{\epsilon_1+\epsilon_2+\epsilon_3}{3}$$

$$\Delta = \frac{\epsilon_1-\epsilon}{\cos 2\theta'}$$

$$\text{TAN } 2\theta' = \frac{\sqrt{3}}{1+2\left(\frac{\epsilon_2-\epsilon_3}{\epsilon_1-\epsilon_3}\right)}$$

Since ϵ and Δ are already defined, a complete solution of the principal strains may be obtained.

$\epsilon_p = \epsilon + \Delta$, $\epsilon_q = \epsilon - \Delta$, and $\Theta = \alpha - \theta^1$

Knowing the principal strains, the principal stresses may be calculated as previously mentioned. The maximum shear stress follows from a known relationship.

$$T_{max} = \frac{S_p - S_q}{2}$$

In our routine work with this mathematical machinery, we have found it expedient to prepare a standard form on which all the necessary operations are performed, starting with three strain readings and ending with the angle of the principal axis, the maximum and minimum principal stresses, and the resultant shear stresses.

The use of such strain deltas has become standard practice for determining stresses in thin sheet. For the general case in aircraft work, such sheets will be operating in the wave state. The additional bending stresses imposed by the buckling of the sheet are surprisingly large. The effect of such large bending stresses render the indications of a delta, attached to one side of the sheet only, very unreliable so far as determining average conditions is concerned. For that reason, unless it is known that buckling will not occur, deltas are invariably installed with gauges back to back. Each leg of the delta is then connected with its counterpart on the opposite side of the sheet to read an average stress which automatically cancels out the additional bending stresses due to wave formation. Fig. 5 which is based on tests conducted to investigate these points, illustrates them very well. Note the wide divergence in shear stress indicated by the surface stresses on one side of the sheet only, by comparison either with the delta on the other side or with the brown overall shear. The actual shear is precisely checked by the back-to-back arrangement. It is startling, sometimes, to discover how very different local conditions may be from what general theory might indicate. In this case, the reasons for the divergence are, of course, understood. The ability of the gauges to indicate local surface conditions has even been used to advantage to check these very stresses in theoretical work being carried out on the development of the tension field theory.

There are cases in which a test laboratory set-up with back-to-back arrangement of strain gauge " rosettes " is not feasible: for instance, one side of the sheet may form the inner wall of an integral fuel tank. It is possible in such a situation to install two deltas on the same side of the sheet, oriented so that each leg is parallel to its counterpart, spaced one-half wave length apart, and connected as before, to read an average strain. The wave length can be calculated for a particular stress with reasonable accuracy ahead of time and tests have indicated that the accuracy is not rapidly affected by slight errors in the spacing.

Tests have substantiated the fact that accuracy is not affected by orientation with respect to the principle stress directions.

There is some gain in accuracy achieved by installing the three gauges forming the delta in as small a pattern as possible. The need for dummy gauges for all except total bending differential applications arises from the fact that, in the bridge circuit, it is necessary to balance the resistance of the gauge unit itself against an identical resistance. The best way to secure an identical resistance is to use another gauge. Furthermore, the most satisfactory temperature compensation is achieved if the extra dummy gauge is mounted on metal having the same composition as the metal on which the stresses are being investigated. It is essential, furthermore, that

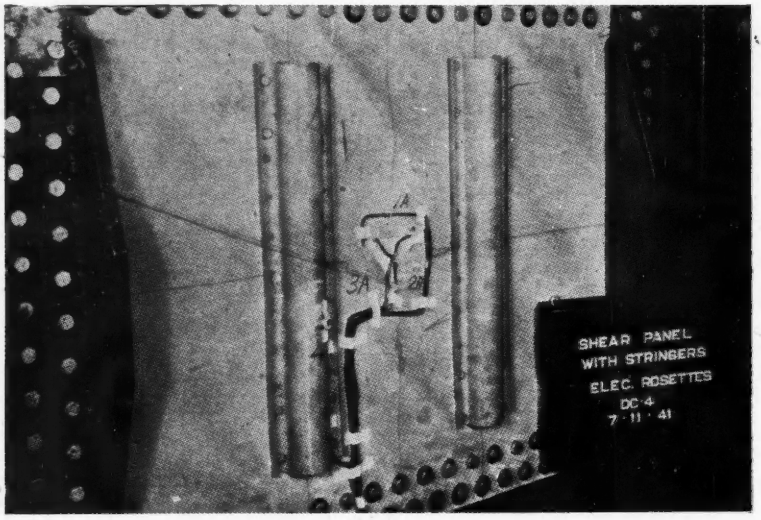


Fig. 4

the extra dummy gauges be mounted on metal which is not itself stressed. The dummy gauges for the strain delta are mounted on a separate piece of metal so supported that stresses cannot be transmitted to it from the web on which it is supported. Such is the typical and essential practice.

Instrumentation Summary.

In summary, we have at our command distinct methods for coping with each type of stress with which we may be concerned. Strain, deflection, and vibration are seen to be brought closer together than ever by virtue of the fact that they can be investigated (simultaneously, if need be) by the same instrumentation.

In what follows, reference is made particularly to the use of methods under discussion, to static tests of complete aeroplanes, or to tests made during flight. In such cases, the wiring between the gauge units and the associated apparatus becomes a considerable undertaking. The gauge units themselves are cheap, but it is necessary to use as few channels as possible to hold the installation wiring problems to a reasonable scale. Furthermore, in flight testing, the number of channels available simultaneously is sharply restricted. None may be wasted on extraneous information. In fact, the larger problems under investigation often must be broken down into phases which can be handled one at a time by the available equipment. Means must then be provided to switch from one phase to another when, as is usually the case, more than one phase must be covered during a single flight.

Figs. 6 and 7 show photographs of the installation of the amplifying and recording apparatus set up during a static test of a complete aeroplane. In this case, cables and all the wiring from all the gauges installed all over the aeroplane have been brought to a stand conveniently set up outside of the aeroplane. The unit to the left in Fig. 6 is a six-channel amplifier with its associated meters. The instrument is a Brown Electronic Potentiometer being used to record indications from some 300 strain-gauge installations. This instrument incorporates an electronic amplifier and a high-speed self-balancing bridge system. A switch panel is shown to the right which permits selection of any one of the 300 channels. For flight-test work, an installation similar to that shown on Figs. 8 and 9 is typical. The rapidly varying nature of the stresses, particularly the short duration of the peak stresses, in which we are most interested, makes it necessary to record the results with some form of recording oscillograph. In our work a twelve-channel Miller oscillograph is used. In this case a twelve-channel amplifier and a twelve-channel Miller oscillograph are shown installed in the XB-19, this equipment being thoroughly suitable for dynamic loads in contradistinction to the two preceding illustrations which are of an installation suitable only for static test work.

It is necessary to diverge from discussion of strain-gauges *per se* for a moment to point out that various other uses may be made of this equipment simultaneously; for instance, very reliable accelerometer pickups, which function as an inductance bridge just as the strain gauges function as resistance bridges, have been developed and may be used simultaneously with the strain gauges. Also, both the accelerometer and the strain-gauge units are ideal vibration pickups, and they interchangeably and simultaneously record vibration frequencies and their relative amplitudes. Moreover, a little ingenuity will make the same instrumentation supply much additional information. For instance, such things may be done as recording pressure variations in a hydraulic line thanks to its expansion and contraction picked up by a strain gauge cemented around the line, or indicating a landing wheel rotational speed by suitable contact points on the wheel, superimposing a

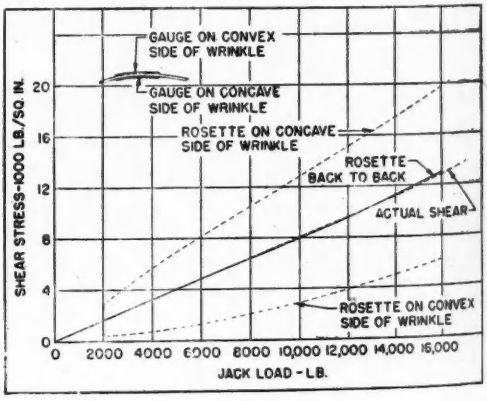


Fig. 5

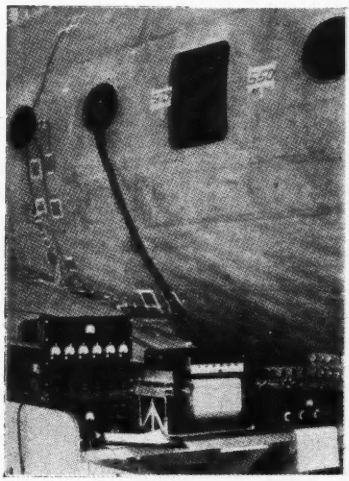


Fig. 6

record on a channel already being used for some other problem as in Fig. 3. Spatial deflections may be measured by suitably forcing the deflection of a simple cantilever beam which is equipped with bending strain gauges and is calibrated in terms of deflection of the beam.

The record from the oscillograph may represent an incredible amount of information, often not obtainable in any other way.

Applications.

As an example, consider the familiar and still difficult problem of the distribution of direct and torsional shear in a multispar stressed skin wing. The shear stress distribution is represented by Fig. 10A.

The stresses shown will be the combination of direct and torsional shears plotted vertically from the contour of the wing section as a base. Their value largely determines the skin thicknesses, the spar web thicknesses, something of the stiffener arrangements involved, and the rivet patterns for seams in the wing covering. Methods of calculating such shear intensities are somewhat involved. A substantiation of their accuracy is desirable. The necessary arrangement of strain deltas is indicated by Fig. 10B. This installation would involve 12 deltas, or 36 channels. So great a number of channels limits such an investigation to static test work rather than flight test.

In calculating such a stress distribution, the usual procedure is to distribute the direct shear among the spars in proportion to the spar web area. Such a procedure is based on the fact that the twisting deflections of a monocoque wing are small.

The air load applied to the wing and acting at its centre of pressure causes a torsional moment about a spanwise axis of the wing. The magnitude of this moment depends on the distance of the selected axis from the air load centre pressure. For convenience, the elastic axis, defined as the locus of points about which no angular deflection would occur were load applied at that point, is the usual reference axis. The corresponding torsional moment is then distributed among the various torque boxes (three in Fig. 10) in proportion to the torsional rigidity of each. Finally, the shear stresses in the wing covering involved in carrying the bending loads to any bending material located between the shear webs, are calculated. All three shear stresses are combined to give the typical pattern of shear-stress intensity of Fig. 10A. This procedure, therefore, involves the determination of the elastic axis position in order to establish the value of the torsional moment.

So far, agreement between mathematical methods of calculating the elastic axis position with test determination has not been satisfactory.

Since the axis cannot be located accurately during the design stage, common practice has been to define

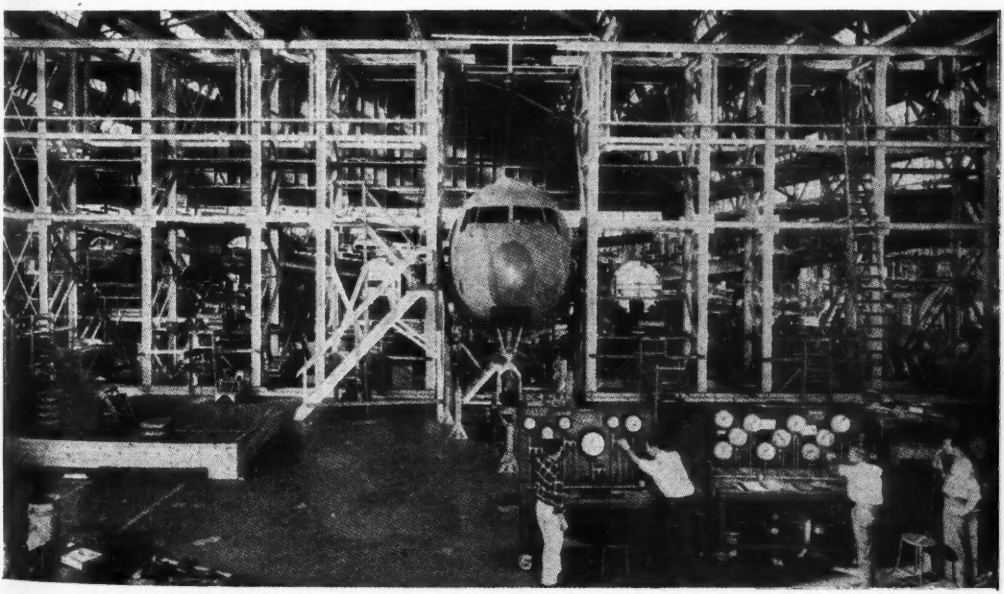


Fig. 7

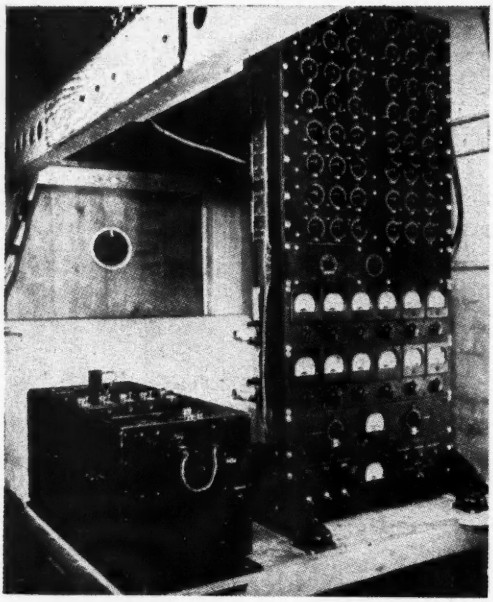


Fig. 8. Front view.

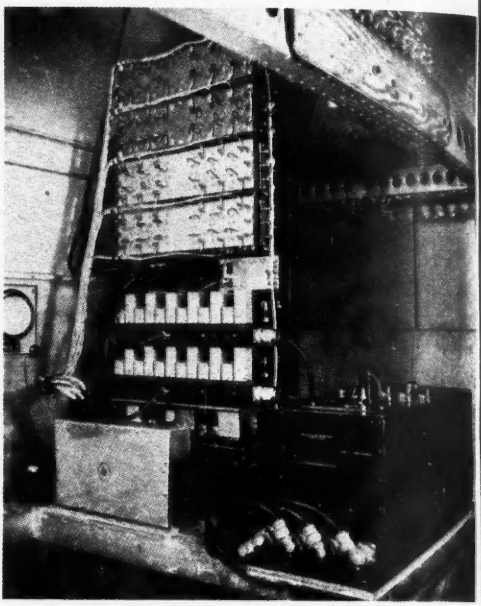


Fig. 9. Rear view.

conservatively a range of chord positions within which the axis could lie and to design the wing structure for the loads implied by the extremities of the assumed range. This method involves a doubling of much analysis work and the inclusion of more structural weight than is actually necessary were the position of the elastic axis accurately known.

Starting from the strain-gauge data derived from an installation as indicated by Fig. 10B, it is possible to work in the opposite direction, separate out the various contributory shear stresses, and arrive at the true shear centre or elastic axis. That is being done, and a wider experience with it will lead ultimately to reduced analysis time and structural weight.

It may be noted in addition that a similar procedure would be very suitable for an investigation of the shear lag problem arising from the shear deflections which correspond to the shear stresses in the wing covering. The ability to determine experimentally the actual shear

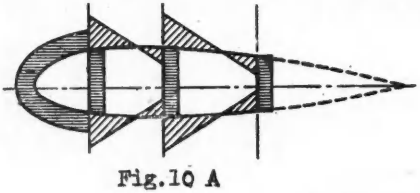


Fig.10 A

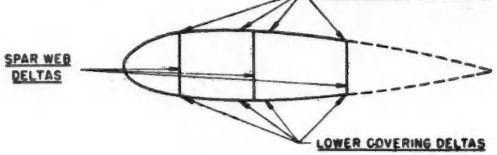


Fig.10 B

distribution by use of a shear delta on the wing covering between each item of bending material should accelerate greatly the development of a mathematical treatment suitable for several engineering applications.

As another sidelight on the use of a similar installation in flight-test work, there recently has been a growing concern over the effects of compressibility upon such items as the aerodynamic twisting moment imposed on the wing at velocities approaching the speed sound. The increases predicted by pure theory have not been evident in flight-test work so far conducted. Actual measurements of the magnitude and, more particularly, of the variation in magnitude with dive speed from a simplified version of such an installation are planned to throw light on this subject.

Another basic problem is the substantiation of the broad assumption that the simple bending theory applies to a beam of the complicated cross-section of a typical multispar cantilever wing. I have mentioned previously the usual assumption that twisting deflections are small, which is to say that each spar assumes the same deflection curve. Presumably, spars are forced to the identical wing deflection curve (as distinct from their individual curves) by the torsional rigidity of the outer covering and the stiffness of the ribs. Actually, neither of these considerations is infinitely effective, and some independent motion of spars does occur. Such independent spar action is only likely to become important where special design problems occur such as discontinuous spar arrangements to provide room for retractable landing gears or opening for bomb bays, and so on. In such cases, the design problem may become very difficult indeed.

Assumptions as to the behaviour of the spars under load must be made, and the resulting structure itself becomes a function of those assumptions. Substantiation of the assumptions that have been made in a particular design are therefore important, both for that design and for the development of good judgment in assumption that must be made for new designs.

At least one very complicated problem of this nature

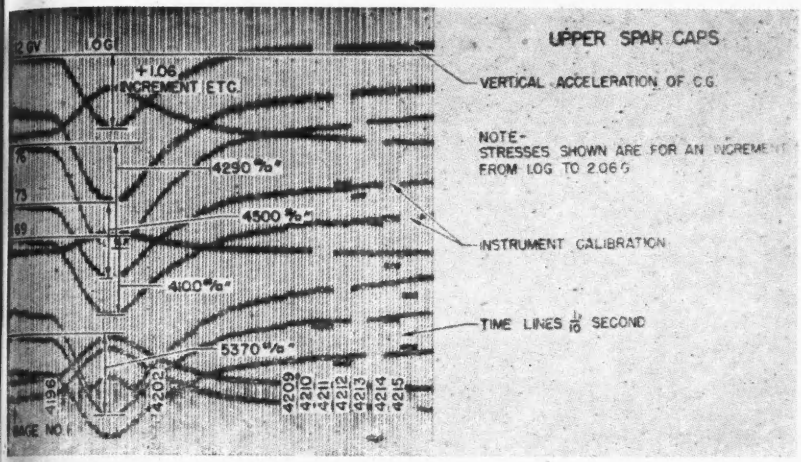


Fig. 11

has been investigated in flight. For that purpose, the state of stress of several critical cross-sections of the wing in the affected area was determined by data from average reading strain-gauge installations on each spar cap and strain deltas on each spar web. By comparison with calculated stresses and readjustment of the original design assumptions, a better understanding of the behaviour of the entire assembly is achieved. This work on an experimental airplane means that a production redesign for the usual conditions of greater power and greater weight would be accomplished with more despatch and precision than would otherwise be possible. This might affect a long-range saving many times greater than the cost of securing the information.

Fig. 11 is a reproduction of a portion of the records obtained during this particular investigation. The traces showing that variation of stress in the upper spar caps at one section of the wing have been marked, together with some auxiliary data. The reproduced portion of the record is typical. From it, the entire history of events throughout a sharp pull-up, one of many made for these test purposes, is determined. The magnitude of the stress at any point and at any instant is indicated by the departure of the trace from its steady flight condition base. Fig. 12 is another part of the data typical of records of strain deltas on the spar webs under the same conditions.

One of the basic problems with which the structural designer is always faced in proportioning the structure of a new design is the nature of the transfer of load between the wing and fuselage at their intersection. In large all-metal designs, for reasons of structural continuity, there is usually in addition to fittings between spars and fuselage frames, a continuous shear connection between wing covering and fuselage skin. If it were not for this shear connection, the magnitude of the reactions at the fittings would be simple problem in statics. With it, however, the state of affairs is indeterminate.

The usual procedure in design is to make overlapping assumptions as to the proportion of the torque load that is carried directly by the wing to fuselage fittings and as to the proportion that is carried by direct shear transfer from the wing covering to the fuselage skin. The problem is of particular importance whenever, for functional reasons, it is desirable to keep the interior fuselage space free of obstructing bulkheads. This arrangement means that loads transferred to the fuselage to the fittings are actually carried by ring frames which are usually shallower in depth than the structural designer would prefer.

While we have come laboriously over a period of years to have a fair understanding of the state of affairs, we still must resort to overlapping assumptions, which phrase is a reassuring way of saying that the design is overly conservative.

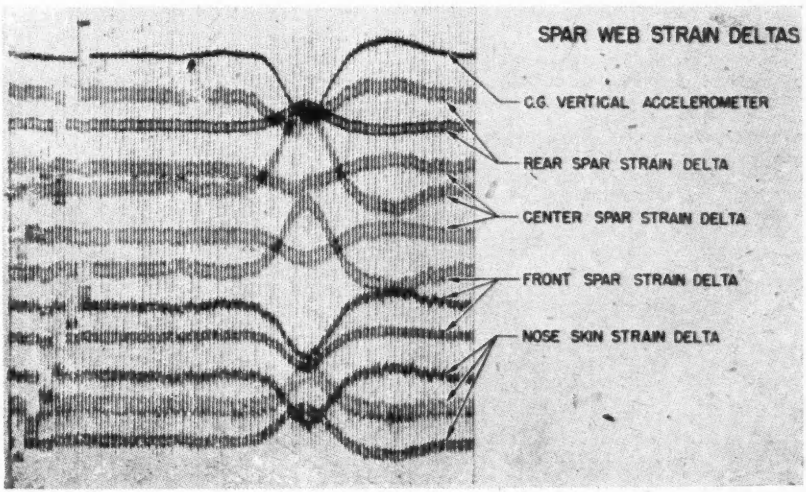


Fig. 12

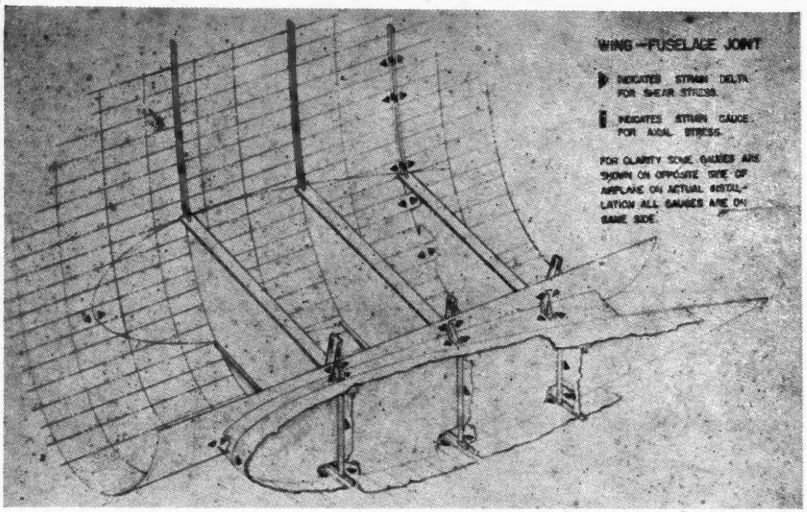

WING—FUSELAGE JOINT
INDICATES STRAIN DELTA FOR SHEAR STRESS.
INDICATES STRAIN GAUGE FOR AXIAL STRESS.
FOR CLARITY SOME GAUGES ARE SHOWN ON OPPOSITE SIDE OF AIRPLANE ON ACTUAL INSTALLATION ALL GAUGES ARE ON SAME SIDE.

Fig. 13

Part of the investigation being covered by the set-up shown in Figs. 6 and 7 was concerned with this problem, shown diagrammatically by Fig. 13. Strain deltas at critical points in the wing shear system just outboard of the fuselage, on the fuselage skin adjacent to the wing intersection, and close to the spar frames, together with a number of gauges sufficient to determine loads being imposed on the spar frames immediately above the wing fittings, permit a precise determination of the nature of the load transfer.

The resulting information, like all similar information, serves to substantiate the design assumptions that have been made; permits more intelligent revision of design as inevitable increases on load-carrying capacity during the future of the particular design is concerned; permits more accurate disposition of shop salvage problems that may arise in the affected area; and, last but not least, permits the reduction, or perhaps the elimination, of the need for overlapping assumptions in the design of the next similar airplane.

It has been previously mentioned that both the strain-gauge and accelerometer pick-ups were suitable for vibration recording work. In fact, they become inseparable in flight-test work for every strain recording channel records any variation in stress that may result from oscillations in the aircraft structure. The feature is sometimes taken advantage of in order to determine the nature and the seriousness of such oscillations as may occur. For instance, Fig. 14 is a section of the oscillograph record from an installation that was set up for the purpose of determining the severity and structural importance of the mild oscillation of the tail that was known to occur in an airplane under investigation. The tail was exited at the frequency with which we were concerned on the ground, and the record of Fig. 14 was made. It was obvious from observation that the amplitude of the oscillations were much greater on the ground than in the flight; nevertheless the stresses so measured were determined and were found to be neligible. An identical record was then made with the airplane in flight while similar oscillations were occurring, and it was possible to demonstrate that the oscillations occurring in flight were much less serious. It was shown also that the stresses resulting from flight oscillations were correspondingly less serious than those which had occurred on the ground, which were, as we have seen, negligible. This was an important point, and one that it might not have been possible to establish so completely by any other means.

Limitations.

So far, I have dealt with the favurable side of the picture. In order to restore a proper sense of proportion, however, it will be necessary to discuss some of the difficulties that are involved. There are, of course, technical difficulties. Low stresses may be indicated if there exists a lag due to imperfect cementing of the gauge to the surface in question. Yield of the wire of the gauge itself at higher stresses may cause difficulties; moreover, troubles common to any involved electrical circuit may be encountered. Such problems, however, properly belong to a discussion of the development of the equipment itself rather than to its application, and reference is made again to the same two papers mentioned earlier.

Aside from technical difficulties with the equipment itself, there are endless human problems involved in any large-scale application of this technique. One definitely cannot expect the mere thoughtless application of huge numbers of gauges to lead automatically to wonderful results. Careful thought and planning ahead of any investigation result in reducing the complication of the installation to a minimum and are utterly essential. Preparation for the rapid, accurate, and permanent recording of pertinent data is necessary.

Such difficulties as do occur may not be so serious in laboratory work, for tests can be repeated there with

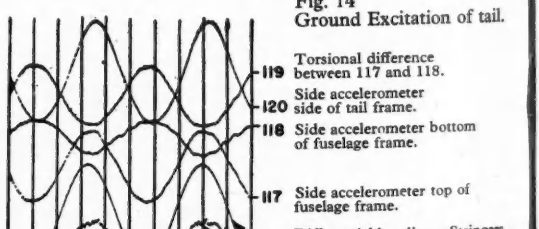

119 Torsional difference between 117 and 118.
120 Side accelerometer side of tail frame.
118 Side accelerometer bottom of fuselage frame.
117 Side accelerometer top of fuselage frame.
116 Differential bending—Stringers opposite sides of fuselage.
115 Side accelerometer—side of fuselage frame.
114 113 112 Strain delta bottom of fuselage.
111 110 109 Strain delta top of fuselage.

Fig. 14
Ground Excitation of tail.

reasonable ease but, in large-scale static test work and in flight-test work, an error that invalidates a recording of events may spoil an opportunity to secure data that will never occur again.

Even apparently satisfactory results, so far as the equipment is concerned, have to be used with a great deal of understanding. It is necessary to have sufficient understanding to recognise misinformation, to analyse the nature of it, to trace the trouble to its source, and to repeat that process until satisfactory results are being achieved.

The most serious problem of all, under the present circumstances, may well be that of limiting these methods to applications reasonably sure to produce directly useful information and not a confusion of unassimilated data.

I have been primarily concerned in this paper with conveying the idea that there are techniques and equipment available, the application of which, though difficult, gives an insight into the problems of structural behaviour, which has never been possible before. Such an insight is particularly important in aircraft work with thin metal structures, the whole theory of which is still in the development stage.

ELECTRICAL MEASUREMENT of HUMIDITY of SPINNING MATERIAL

By H. MAHLO. (From *Zeitschrift des Vereins Deutscher Ingenieure* (VDI), Vol. 85, Nos. 49–50, Dec. 13th, 1941, p. 961.)

[The article is an abstract of two publications of the author: "Electrical measurements of the Humidity of Cellulose Wool and other Textile Fibres," Heidelberg, 1940, 97 pages and 66 Figs., and a Lecture to the Textile Conference, Dresden, 1941.]

A METHOD based on a drying process, called conditioning, has long been used for measuring the humidity of spinning material. When applied carefully this method can determine very accurately the humidity of samples weighing several hundred grams. But the method fails if great quantities of material of variable humidity are to be tested. If a sample weighing several hundred grams is taken from a cellulose wool bale in which the humidity varies, as is usual by as much as 5 per cent, the humidity of the sample cannot be regarded as the mean value of the humidity of the whole bale. If the accuracy is improved by measuring a great number of samples, the method becomes too costly.

Possibilities of Electrical Measurement.

P. Lippke has proposed a method for a continuous measurement of the humidity by determining the variations of the capacity and of the dielectric losses of a stream of cellulose wool coming out of a dryer and led through a condenser. The mean humidity of a bale may be found by determining the mean values of the readings or by using an integrating measuring instrument.

It is evident that it is impossible by any electrical method to measure the humidity directly. Only electrical qualities that are influenced by the humidity, such as the ohmic resistance, the dielectric constant or the dielectric losses can be measured.

Researches made by Mahlo, based on the results of previous researches by other investigators, led to the following results: It was confirmed that resistance and humidity are connected by an exponential relation. Some materials show a sudden change in the curvature of the characteristic line depending on the kind of linkage of the humidity with the material. The dielectric constant is a linear function of the humidity. But the diagrams show also a bend in a certain range. Curves showing the relation between the dielectric losses and the humidity have no uniform character. They have exponential character for low humidity contents and become straight lines for high humidity.

Calibration curves have to be used for all these methods of measuring. A special curve for each material has to be used. These curves may be influenced by changes in the cleaning or oiling process of the manufacture of the cellulose wool. Temperature of the fibre and pressure have some influence on the results of all measuring methods. Since the calibration curve is logarithmic, very small variations of the humidity can be determined by the measurement of the resistance. But great errors result with this method, if the humidity is not uniform over the cross section used for the measurement. In such a case the result of the measurement corresponds practically to the part with the highest humidity. The measurement of the dielectric constant is more accurate in consequence of the linear character of the calibration curve. Even when the humidity is not uniform throughout the part used for the measurement, correct mean values result so long as the variations of the humidity are not too great. The error arising from the measurement of the dielectric losses in a cross section of unequal humidity is greater than the error arising from the measurement of the dielectric constant, but lower than the error arising from the measurement of the resistance.

New Measuring Methods.

Two new methods for the measurement of the humidity of complete cellulose wool bales have been developed in the laboratories of the Süddeutsche Zellwolle A.G., Kehlheim/Donau. They are based on the results of the researches mentioned above.

One method uses the measurement of resistances. Two pairs of contact needles of 18 cm. length, fixed in a handle so as to be 4 cm. apart, are thrust into the cellulose wool bale. The resistance between the pairs of needles is measured. Such measurement is made for 10 different points of the bale, and the mean value is derived. The method is suitable for high grades of humidity. For a humidity lower than 10 per cent. the accuracy is poor. The scale of the measuring instrument may be calibrated in per cent. humidity. Differences in the staple and the titre of the wool have no influence on the calibration, if other qualities of the material are not changed. The test of one bale takes six to eight minutes, when 10 measurements are made.

If the humidity is uniform, the results have little deviation (0.2 per cent.). With bales of unequal humidity the deviation is greater (0.6 per cent.). The accuracy may be raised by making more measurements per bale. The method has been used for eight months as a check against too high humidity of bales and works satisfactorily. The method has the advantage that it gives an information about the variation of the humidity throughout the bale. The instrument is very simple, and may be used for all sizes of bales. The measurements enable the material of bales of different humidity to be mixed so as to yield a mixture of mean constant humidity.

A second method of measuring is based on the measurement of the dielectric constant. The bale is placed between two condenser plates enclosed in a wire cage and the capacity is measured. The results of this method were compared with those of the conditioning method and of the measurements of the resistance. Good agreement was found. The comparison of the values found for the humidity by the different methods was difficult in consequence of the considerable deviation of the results of the conditioning measurement. These comparative tests will be continued. But the results obtained so far by the measurement of capacity suggest that this method is very suitable for finding the mean value of the humidity of a bale quickly and with sufficient accuracy.

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NEW EQUIPMENT

C.V.M. 65 VERTICAL MILLING MACHINE.

THE illustration shows the latest design of C.V.M. 65 milling machine manufactured by Messrs. Kendall & Gent (1920) Ltd., Victoria Works, Gorton, Manchester. It is similar in design to the well-known C.V.M. 40, but its capacity is larger, and the spindle motor is vertically mounted on the spindle head.

The major castings are rugged in design with sliding surfaces of densified iron; the spindle runs in combination ball and roller bearings housed in large diameter densified iron sleeve; the gears are of high tensile nickel chrome steel and slide on splined shafts. All drives are effectively short due to endshield mounted motors; the speed and feed selections are particularly simple and easy; and comfortable control has received special attention.

The SPINDLE HEAD has a speed range varying from 10 to 284 r.p.m. Nine spindle speeds are obtained directly from a single control lever, the speeds being in progression of cutter diameters. On the opposite side of the head standard pick off gears give surface speeds from 30 to 80 ft. per minute.

The head has the drive motor vertically endshield mounted and is elevated by power under push button control. Movement is limited in a downwards direction by adjustable dead stop and upwards by safety limit switch.

The 6" of hand adjustment is by large diameter handwheel fitted with accurate graduated dial. Movement is effected by worm with incorporated anti-backlash device. Dial indicator and adjustable stop for setting the cut is provided. Spindle nose has No. 6 Morse taper cone and B.S.I. mounting for large face cutters.

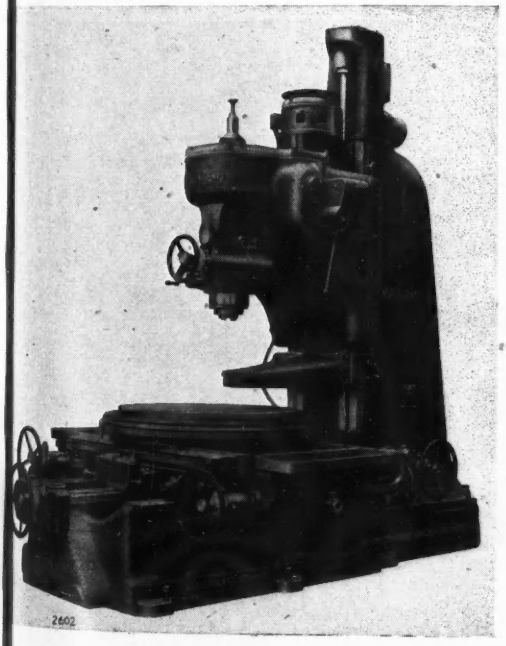
The BASE is particularly massive and has its main supporting surfaces directly under the spindle. Two well spread auxiliary sliding surfaces ensure the necessary stability of table. Ways have a high degree of hardness and are protected by sliding covers at the rear.

The COLUMN carrying the milling head and elevating mechanism also has densified ways. It is secured both horizontally and vertically to the base. Electrical gear is built into the column and has automatic cut out connected with the inspection door.

FEEDS are selected by direct reading dial change operated by cross handle. Sliding gears are of high tensile nickel chrome and run in an oil bath. The box is fitted directly under the left hand end of the table. Direction of traverse is electrically selected.

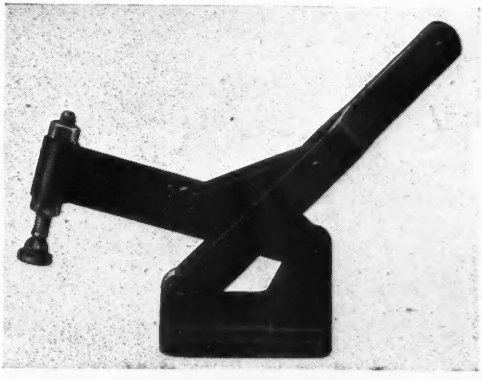
TABLE. The machine is fitted with circular table, with horizontal and transverse traverse in addition to the circular feeds.

The EQUIPMENT consists of cutter drawbolt, grease gun, tool kit, dial indicator, depth stop, electric coolant pump and fittings and lighting fixture. Swinging stay in photograph is available as an extra.



DEVELOPMENTS IN "SPEETOG" CLAMPS.

A NEW "SPEETOG" quick-acting toggle clamp is announced by Messrs. Speed Tools Limited, 10-16 Rathbone Street, London, W.1. It is equipped with a ball and socket plate instead of the normal plain holding-down screw, and has been introduced to meet the demands of the Aircraft Industry.



It is claimed that the new "SPEETOG" clamp has a wide field of use for clamping operations on duralumin sheet, on wood and on other materials which would be liable to bruise if clamped with the holding-down screw of normal clamps.

An illustration of the new "SPEETOG" clamp, known as Model 1A, is given on this page. The holding-down plate, one inch diameter, is arranged to swivel around the nose of the clamping screw. The clamping screw carrying the swivel plate can be adjusted to accommodate different thicknesses of materials. The swivel plate is serrated on its underside so that a piece of felt or other soft material can be cemented on to it to provide additional safeguard where it is desired to avoid marking the materials clamped.

Model 1A as illustrated is $12\frac{1}{2}$" high overall, has a base 5" by $3\frac{1}{8}$", throat size $3\frac{1}{4}$", and the radius of the clamping arm is 8".

CATALOGUES RECEIVED

Carbon in Steel Determination Apparatus.—A 4-page leaflet issued by Messrs. Griffin & Tatlock Ltd., Scientific Instrument Makers, Kemble Street, London, W.C.2, describes an improved Ströhlein type apparatus for use in the accurate estimation of carbon in steel to 0.01 per cent. in about one minute. The apparatus is based on the volumetric method. The sample of drillings is burned in a stream of oxygen, the resulting CO_2 and excess oxygen are collected in a special jacketed burette and the CO_2 then absorbed. On passing the excess oxygen back to the burette the diminution in volume is read against a scale calibrated directly in percentage of carbon. Copies of the leaflet obtainable upon request.

Symons Sand Screens. Bulletin No. 89 of the Nordberg Manufacturing Co. Ltd., Bush House, London, W.C.2, deals with the Symons Sand Screen, which has been designed specially for removing oversize, or undersize, foreign matter from fine dry materials such as sands, chemical powders, &c. The Bulletin, copies of which may be obtained free of charge, gives detailed information on this high capacity screen with easily variable vibration.

BOOKS RECEIVED.

American Standard Definitions of Electrical Terms, C42. A NEW American Standard known as Definitions of Electrical Terms, C42, sponsored by the American Institute of Electrical Engineers, is now ready for general distribution. It is the first time the definitions of the important terms common to all branches of the electrical art as well as those specifically related to each of the various branches have been assembled and printed under one cover.

This glossary is the result of more than twelve years' work of a sectional committee of 46 members having 18 subcommittees drawn from available specialists. More than 300 individuals have given material assistance and many others have assisted in specific instances. The 34 organizations represented on this sectional committee include the national engineering, scientific and professional societies, trade associations, government departments and miscellaneous groups.

The price of the book is $1.25 (outside U.S.A.), and it may be obtained by A.I.E.E. Headquarters, 33 West 39th St., New York, N.Y.

Economic Control of Iron and Steel Works. By F. L. Meyenberg, M.I.Mech.E. Chapman & Hall Ltd., London, 1942. Cloth, $5\frac{1}{2} \times 8\frac{1}{2}$ in., 332 pp., 25s. This is an excellent book and it should prove to be of great use to executives in the Iron and Steel industry, and also in other industrial works. By stimulating better co-operation between commercial and production departments, the author has contributed to greater efficiency in works where consideration would be given his suggestions. In a Foreword by W. I. Larke, K.B.E., Director of the British Iron and Steel Federation, the book is strongly recommended to all those already in, or aspiring to, executive positions in industry, whether technical, productive, commercial or financial.

Machine Shop Yearbook and Production Engineers' Manual, 1942. Editor: H. C. Town, M.I.Mech.E., M.I.P.E., Foreword by Sir Alfred Herbert, K.B.E. Paul Elek Publications, London. Cloth, $5\frac{1}{2} \times 8\frac{1}{2}$ in., 558 pp., 25s. The aim of this book is to present "a descriptive topical review of established practice, and of the research and development work carried out during the previous 12 months, as recorded in the transactions of the numerous technical societies and in the technical press throughout the world." The first part of the book consists of eight articles from which those on Machineability by H. Geo. Schlesinger, and on Pitch testing of gears by W. A. Tuplin, D.Sc., are of especial interest and value. The reader will find, in the second and third part of the book, an interesting account of workshop practice and of modern developments of workshop equipment as manufactured by well-known engineering firms in this country.

Practical Mathematics. By Louis Toft, M.Sc., and A. D. D. McKay, M.A. Sir Isaac Pitman & Sons, Ltd., London, May, 1942. Cloth, $5\frac{1}{2} \times 8\frac{1}{2}$ in., 612 pp., 15s. The book is printed on thin paper and is very pleasant to handle. It contains in one volume the whole of the mathematics included in the syllabus of the final examination in this subject for the London University degree in engineering. The authors have well succeeded in treating the subject on practical lines, and in offering the students a working knowledge of the subject rather than a knowledge of the strictly logical development of each of its branches.

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